Final Report *March 1969*

SHELTER CONFIGURATION: ENVIRONMENTAL CONTROL SYSTEMS AND RELATED PARAMETERS

By: FRANK C. ALLEN

Prepared for:

OFFICE OF CIVIL DEFENSE
OFFICE OF THE SECRETARY OF THE ARMY
WASHINGTON, D.C. 20310

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OCD Work Unit 1236A

STANFORD RESEARCH INSTITUTE
Menlo Park, California 94025 · U.S.A.

115

Summary of
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Approved by:

H. L. DIXON
Executive Director
Urban and Social Systems Division

SUMMARY

Introduction

It is imperative that heat and moisture produced by people and other sources be removed from a shelter to maintain a tolerable thermal environment. If atmospheric conditions approach the limiting environmental state, large quantities of ambient air are required to perform this control function with a simple ventilation system. Ambient air conditions that would probably occur in the warmest season then become the most critical consideration in determining the necessary capacity of a particular ventilating system.

Per capita ventilating rates required in any region of the United States have been determined on the basis of maintaining an adjusted, daily average, effective temperature of 82° FET or less during 90 percent of the days in a normal year, with the premise that the thermal environment in a shelter is spatially uniform. Typical systems devised for ventilation of identified fallout shelters utilized manually-powered fans that exhaust air from occupied spaces to the atmosphere through plastic ducts. In general, one or more of these fan units are located in areas that are remote from openings through which outside air must enter the shelter, and air flows through occupied spaces toward the exhaust fans. Thus, the cost of a system of rigid distribution ductwork is avoided.

A ventilating system that performs satisfactorily during warm humid weather does not necessarily perform satisfactorily in a cool or cold season. Each person in a shelter is a source of heat and moisture that induces weak, localized currents of recirculating air in which velocity components tend to be vertical. If the ventilating air is distributed in the space in proportion to the distribution of people, it is possible to obtain a substantially uniform environment. If the ventilating air is introduced at one end of the shelter or at the periphery of a group of people, it is quite apparent that the resultant velocity vectors will have horizontal components in the direction of air flow, at low as well as high rates of ventilation. Temperature and humidity of the ventilating air will then increase progressively as the air moves through the space; nonuniformity of the thermal environment is characteristic of this system arrangement. Since the metabolic response of individuals depends largely upon air temperature in the immediate vicinity, the proportion of sensible heat added to the air will usually decrease and the proportion of latent heat or

moisture added will increase as the air passes through the populated space. The characteristic process line will be quite different from the linear mixing process associated with a uniform environment. Under conditions for which the ventilating rate is determined, persons at the leaving-air end of the shelter space will be exposed to the design effective temperature, but persons at the entering-air end will be exposed to entering air conditions. If there is no recirculation or conditioning of the air, the entering state will be essentially the same as the state of atmospheric air. In a warm season, people in the stream of entering air would be most fortunate; during cooler weather they would need appropriate clothing; during subfreezing weather, many individuals would become distressed. even though clothed in heavy garments.

It is the purpose of this set of studies to investigate and resolve problems associated with nonuniform shelter environments, to devise economical shelter-system configurations that would function satisfactorily in all seasons, to determine performance characteristics of these configurations, and to evaluate system and overall cost-effectiveness. This summary relates to the initial report, which is concerned with the development of basic shelter-system configurations for use as models, with the derivation of analytical rationale for nonuniform environments, and with the evaluation of needed parametric data that are valid over an extended range of conditions. The shelters are intended for protection against primary and secondary effects of nuclear weapons in the 10 to 20 psi overpressure range.

Configuration of Model Shelters

The basic plan developed for use as a model in subsequent studies of system performance and cost-effectiveness is shown in Figure S-1. This plan is a composite that illustrates salient features of three variant shelter types. Each type has a central bay that contains an assembly room and rooms for equipment and utility services, as shown. The Type A shelter would have a rectangular dormitory room along each side of the central bay; the Type B shelter would have a set of three parallel dormitory galleries on each side of the central bay; and the Type AB shelter would have a rectangular dormitory room along one side and a set of three dormitory galleries on the other side of the central bay. Thus, the plan as shown represents the Type AB shelter. N[illegible] capacities and other data relating to the three shelter types are shown in Table S-1. These personnel capacities are based upon equal numbers of people in assembly or active areas and in berthing or passive areas, that is, each person would spend half of each day in active areas and half in passive areas. Alternative management schedules could be devised. The rectangular dormitory room or rooms provide some flexibility in capacity; bunks may have two, three, or four decks. In Table S-1, the number of people in berthing and assembly areas are kept

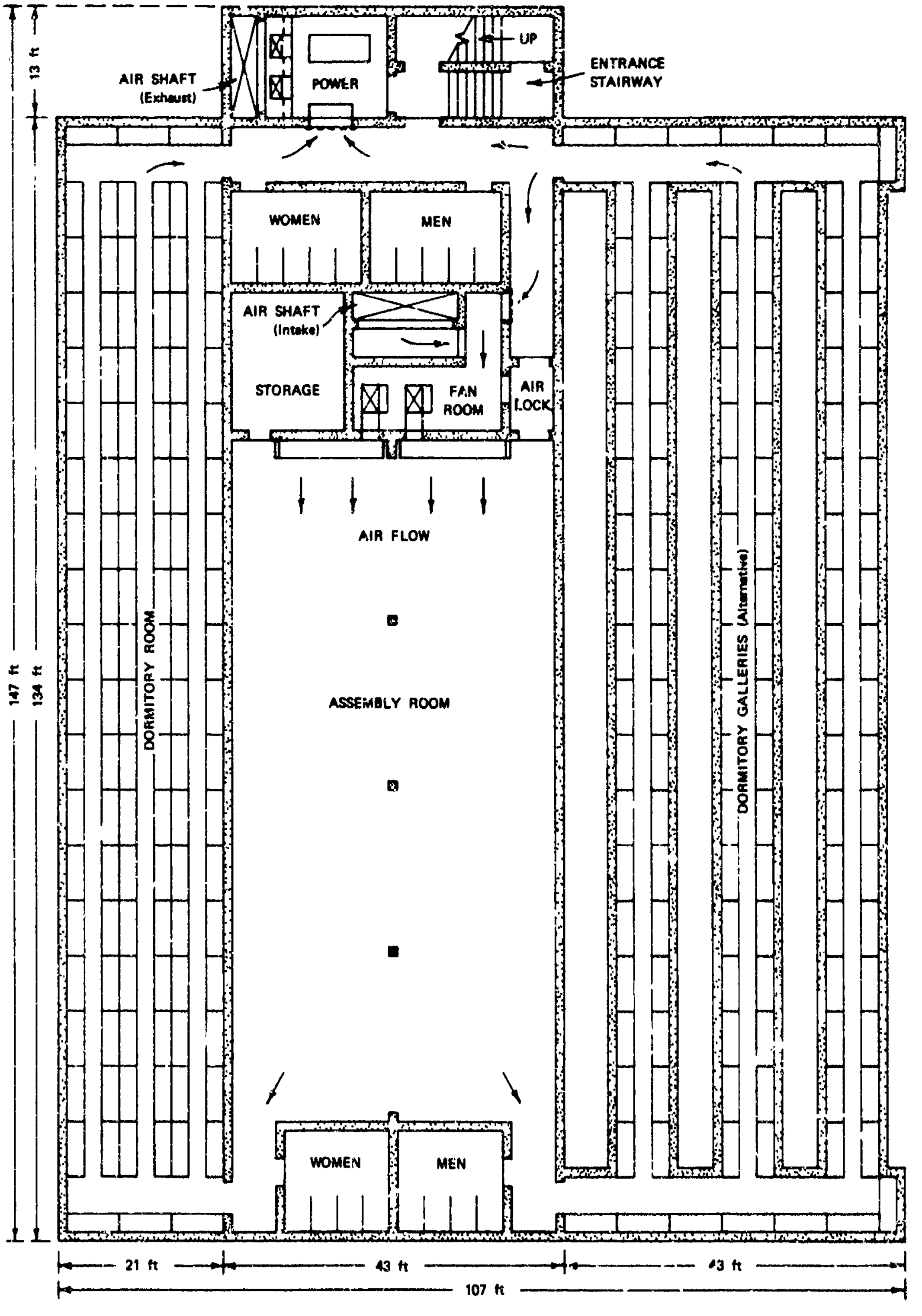

FIGURE S-1 BASIC PLAN FOR MODEL SHELTERS (Composite of Rectangular and Gallery Types)

TABLE S-1

DATA ASSOCIATED WITH MODEL SHELTERS

TYPE OF SHELTER *	HEIGHT OF BUNKS (Decks)		NOMINAL CAPACITIES (Number of People)			UNIT FLOOR AREA **	EFFECTIVE SURFACE CONTIGUOUS TO EARTH
	IN DORMITORY ROOMS	IN DORMITORY GALLERIES	IN BERTHING AREAS	IN ASSEMBLY AREAS	TOTAL IN SHELTER	SqFt/Person	SqFt/Person
A	2	--	456	456	912	9.27	23.41
	3	--	558	558	1116	7.57	19.13
	4	--	648	648	1296	6.52	16.47
B	--	2	488	488	976	9.25	32.02
AB	2	2	472	472	944	9.28	27.86
	3	2	523	523	1046	8.36	25.14
	4	2	568	568	1136	7.69	23.15

* Type A: Rectangular dormitory room on each side of ce ral bay.
Type B: Dormitory galleries on each side of central bay.
Type AB: Composite plan with dormitory room and dormitory galleries.
** Based on total floor area of assembly room and berthing spaces only.

equal by converting passive space to active space, when bunks having three or four decks are used. In all cases, however, the per capita floor area in active areas is about 7.0 square feet per person, which corresponds approximately to the value 7.26 square feet per person in a dormitory room with triple-deck bunks. Bunks having more than two decks are not practicable in dormitory galleries because these galleries would be constructed with corrugated steel or precast concrete pipe having an inside diameter of about 8.0 feet. Moreover, reduction of the per capita area of boundary surface tends to defeat a major advantage ascribed to gallery spaces--the stabilizing effects on the thermal environment to be derived from transient heat conduction phenomena in continguous masses of earth.

The systems for environmental control include the following features:

1. Air intake and exhaust facilities are located near the same end of the structure, with the air exhaust in the room for auxiliary power equipment, the air intake in the room for ventilating or conditioning apparatus and the fans adjacent to the assembly room.

2. Distribution ductwork is virtually eliminated by using advantageous compartmentation and arrangement of corridors and openings to direct the flow of air quite uniformly through occupied spaces.

3. The path for air flow is folded in such a way that used air is returned to the end of the shelter in which equipment is located. This facilitates the use of dampers either to exhaust most of the return air to the atmosphere through the auxiliary power room in warm weather or to recirculate most of the return air through

occupied spaces during cold weather. This eliminates the need for return air ductwork. Recirculation of a mixture of warm return air with cold outside air would obviously reduce the chilling effect of introducing untempered cold air into occupied spaces.

4. The air intake facility incorporates a rock grille that can be wet with recirculated water, well water, or chilled water to supplement ventilating air in the removal of objectionable heat and moisture from the shelter. The cooling effect developed by a wetted rock grille would usually permit a substantial reduction in the required quantity of outside air. A similar rock grille in the air exhaust shaft would provide cooling water for an auxiliary power source. To minimize the possibility of damage to equipment or injury to personnel, provisions are made for the installation of blast valves in both air intake and exhaust shafts.

Environmental Analysis and Associated Parameters

To accommodate conditions that would probably prevail in and around a shelter during any season, a model for environmental analysis should consider the effects of variations in metabolic and other heat loads along the path of air flow. Such a model is shown diagrammatically in Figure S-2. This basic model provides for recirculation of relatively warm air from occupied spaces during cool or cold weather and is specifically applicable to shelters having minimal ductwork for distribution of air. Flow rates of dry air at various points in the system are denoted by the quantities, m_a. The quantities, q_E and q_Z are respectively the per capita lighting load and the variable heat flux over a unit area of boundary surface. A comprehensive treatment of this model involves the simultaneous solution of the psychrometric and heat conduction problems, which have an interface at boundary surfaces. For cases in which the effects of transient heat conduction are significant, numerical solution practically requires the use of an iterative approach with a computer. The effects of transient heat conduction tend to be significant when values are relatively large for the following quantities:

1. Per capita area of boundary surface contiguous to masses of earth or construction materials.

2. Initial temperature differences between masses of materials and the limiting environmental temperature.

3. Product of material properties--thermal conductivity, density and specific heat.

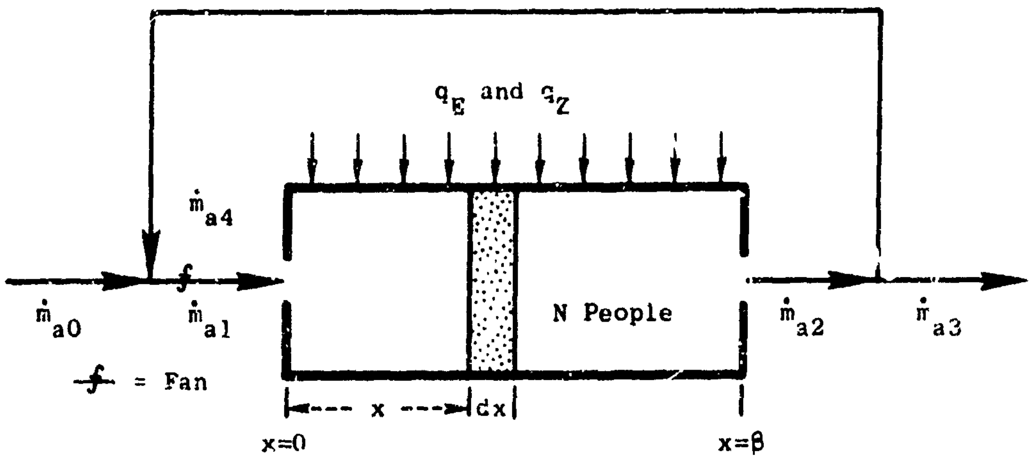

Figure S-2

VENTILATION OF SHELTER SPACE
WITH FRESH AND RECIRCULATED AIR

4. Film conductance at boundary surfaces, for which large values imply appreciable air velocities.

With reference to the model shown in Figure S-2, analytical equations were derived in symbolic form for the air mixing process; for contributions to internal heat and moisture loads made by human metabolism, lighting and other sensible heat sources, and heat flux at boundary surfaces; for differential changes in specific enthalpy and humidity ratio of air traversing the occupied space; for slope of the process line associated with a non-uniform environment; and for psychrometric relationships among properties of moist air. The analysis is carried to the point at which continuation requires the introduction of a rational expression for variations in latent heat loss from the human body as a function of room temperature; an expression that represents metabolic responses over the probable range of shelter temperatures for a standard person dressed in optimum clothing. In this connotation, "optimum" implies that the clothing is appropriate under prevailing conditions at any point in time or space, and that thermal equilibrium will be maintained at all temperatures that would probably occur in a shelter during any season. One such expression is shown in Figure 11 of the main report, but a more representative metabolic equation is envisioned for use in the analysis. Closed analytical solutions appear to be practicable for cases involving steady state heat flux at boundary surfaces as well as metabolic and lighting loads.

The psychrometric analysis is based on the equations of state for ideal gases, and is therefore conveniently applicable to shelters at elevations other than sea level. However, some of the parameters and approximations needed in this type of analysis were evaluated or examined to determine the

magnitude and source of inherent deviations. By means of a curve fitting computer program, polynomial approximations were obtained for the specific enthalpy of saturated water vapor as a function of temperature, for the pressure of saturated water vapor as a function of temperature, and for temperature as a function of saturated vapor pressure. These polynomial approximations were based on tabulated data from the Goff-Gratch-ASHRAE compilation, which covers the entire range of temperatures that are compatible with human habitation.

Since appreciable concentrations of carbon dioxide could develop in shelters that have low rates of air replacement, the effects of carbon dioxide on apparent molecular weights of air mixtures were determined and related graphically to partial pressure of water vapor, humidity ratio, and dew point temperature. The presence of carbon dioxide in appreciable concentrations could influence a thermal analysis more than deviations caused by real gas deviations from ideal gas relationships.

Volumetric deviations for moist air were derived from the basic equation of state and tabulated for comparison. Deviations in specific enthalpy were determined from data on superheated water vapor tabulated in the ASME Steam Tables - 1967. Within the range of habitable environments, both volumetric and enthalpy deviations are relatively insignificant.

Two empirical equations that relate vapor pressure to dry-bulb and wet-bulb temperatures were evaluated with data calculated from thermodynamic wet-bulb temperature relationships. In the tabulated comparison, the partial pressure is more accurately determined by the "Carrier" equation than by the "Ferrel" equation. Two alternative methods for determining vapor pressure from humidity ratio were also compared.

By equating the Stefan-Boltzmann and Newton laws of thermal radiation, a cubic equation was derived for determining equivalent radiation coefficients. This equation was then used to prepare a chart for evaluating these coefficients in either metric or British units, when surface temperatures are known. This chart facilitates estimation of the relative amounts of sensible heat transferred by radiation and convection between body and enclosure surfaces. These equivalent radiation coefficients depend on temperature level and temperature difference, but coefficients for a free convection process depend largely on air velocities induced by temperature and vapor pressure gradients. Under calm conditions in a shelter, induced air velocities and free convection coefficients would be relatively small, and radiation becomes the dominant process for transfer of sensible heat. Similar charts for free and forced convection coefficients would be useful in estimating the effects of various surface temperatures on metabolic losses.

Environmental Criteria

The effective temperature (ET) index has long been used as the basis for environmental criteria, but it should not be considered very definitive. In its usual form, this index applies to normally clothed persons at rest in an enclosure having equal air and wall temperatures. Although the index is capable of correcting for effects of air velocity, references appear to be most often made to the "still" air version. The index was derived from subjective experiments in which exposure times were too short for attainment of thermal equilibrium, and discrepancies encountered in practice have been ascribed to this cause. Applicability of the index to situations that differ appreciably from conditions under which the formulative data were obtained is subject to interpretation. There is a counterpart index for persons at rest but stripped to the waist, which is more representative of conditions that would prevail in shelters during the warm season. Lines of equivalent states obtained with the two versions of the ET index are not parallel; use of the "normally clothed" version is conservative. The counterpart is not in common use, and statements regarding effective temperature customarily refer to the "normally clothed" version. As a result of more recent work, new lines of equivalent states or optimum comfort have been proposed as a nucleus for modification of the index. These lines, most of which are in the zones of moderate temperatures and comfort, suggest a lesser response to changes in vapor pressure than comparable ET lines.

An alternative predictive scheme for evaluating shelter environments is the relative strain index developed in the Division of Occupational Health, Public Health Service. The original formulation was based on persons dressed in normal but damp clothing, a metabolic rate of 100 (kcal)/(hr)(sq m) or 38.9 (Btu)/(hr) (sq ft), which is considerably higher than metabolic rates anticipated in shelters, and an air velocity of 100 ft/min, plus 110 ft/min induced by activity. In general, the relative strain index seems to correct some of the discrepancies attributed to the ET index.

For use in the analysis of shelter environments, the "still" air effective temperature index was converted from dry-bulb and wet-bulb temperature coordinates to dry-bulb and vapor pressure coordinates. Slope coefficients for lines of constant ET are shown in Figure 10 of the main report. These slope coefficients vary linearly in the region from 75 to 90° FET, which is the region for evaporative regulation of body temperature.

In connection with shelter criteria and environmental analysis, major uncertainties relate to the evaluation of parameters for physiological responses, heat stress indexes and metabolic heat transfer.

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OCD Review Notice

This report has been reviewed in the Office of Civil Defense and approved for publication. Approval does not signify that the contents necessarily reflect the views and policies of the office of Civil Defense.

ABSTRACT

This investigation is preliminary to parametric studies concerned with (1) separate and combined effects of all factors that significantly affect costs and performance of environmental control systems for shelters and (2) reciprocal effects of structure and system configuration on overall cost-effectiveness. Representative configurations are outlined for shelters to be used as models in subsequent studies, which are oriented toward underground shelters that would provide protection from weapons effects in the 10-20 psi overpressure range. These configurations provide for recirculation and distribution of ventilating air with minimal ductwork. Alternative dormitory spaces of rectangular and gallery shapes are included; underground pipe galleries tend to enhance cooling effects due to heat conduction in contiguous earth. A concept is shown for a versatile air intake or exhaust facility incorporating a rock grille that could be wet with recirculated, well, or chilled water. A fundamental rationale is developed for analysis of variable shelter environments, and deviations associated with use of ideal gas relationships, polynomial approximations, empirical equations, and variations in carbon dioxide concentration are evaluated. Derived values for equivalent thermal radiation coefficients are shown graphically. Environmental indexes, criteria, and metabolic parameters are examined with regard to adequacy and applicability to a concept for optimum clothing in shelters, and immediate needs for more definitive information are identified.

CONTENTS

ILLUSTRATIONS

TABLES

NOTATION

The following definitions apply to symbolic notation used in this report. In general, quantities can be evaluated in any consistent system of units. The abbreviations, lbm and lbf, are used when appropriate to distinguish between pounds mass and pounds force.

A, B, C . . .	and a, b, c are constant coefficients for numerical evaluation in polynomial or exponential approximations that correlate tabulated or empirical data.
A_F	Floor area of shelter space
A_Z	Peripheral area of shelter boundary surface
A_ϵ	Area of radiating surface in an enclosure
A_o	Area of radiating object in an enclosure
dA_1	Differential element of surface, A_Z
c_p	Specific heat at constant pressure
E	Emissivity factor for radiating object
F(t)	General function relating latent metabolic heat to dry-bulb temperature
$G_1(\Delta t, \Delta p_w)$	General function of temperature and vapor pressure differences relating to total heat transfer at an element, $z_1 dx$, of boundary surface
$g_1(\Delta p_w)$	General function of vapor pressure difference relating to moisture exchange at an element, $z_1 dx$, of boundary surface
H	Total enthalpy of moist air flowing in unit time
h	Specific enthalpy of moist air on a dry air basis

h_a	Specific enthalpy of dry air fraction
h_g	Specific enthalpy of saturated water vapor
h_{fg}	Specific enthalpy of vaporization for water
h_f	Specific enthalpy of water
$h_i \approx h_f$	Specific enthalpy of water condensed from moist air
h_{gd}	Specific enthalpy of saturated water vapor at dew point temperature
h_u	Specific enthalpy of superheated water vapor
dh_M, dh_E, dh_Z	Differential changes in specific enthalpy of air due respectively to metabolic, electrical and surface effects
Δh	Incremental change in specific enthalpy of moist air
h_r	Equivalent coefficient for thermal radiation from a black object in an enclosure
M	Apparent molecular weight of moist air
M_a	Apparent molecular weight of dry air
M_w	Molecular weight of water
M^*	A function of moist air properties at the thermodynamic wet-bulb temperature
m	Mass of a typical sample of moist air. When an analysis is concerned with rates of fluid flow, the sample represents the mass, $\dot{m}$, of moist air flowing in unit time
m_a	Mass of dry air in the sample of moist air
m_w	Mass of water vapor in the sample of moist air
$\bar{m}_{wL}$	Per capita rate of moisture evaporation associated with latent metabol heat

$\bar{m}_{wZ}$	Net rate of moisture condensation or evaporation at a unit area of shelter boundary surface
$\Sigma \bar{m}_{w\Delta}$	Total rate of moisture transfer to air flowing through shelter space
N	Number of people in shelter
n	Number of mass moles in sample of moist air
n_a	Number of mass moles of dry air in sample
n_w	Number of mass moles of water vapor in sample
P	Natural logarithm of saturated vapor pressure, p_{ws}
P_E	Wattage of lighting load per unit of floor area
p	Total or barometric pressure
p_a	Partial pressure of dry air component
p_w	Partial pressure of water vapor
Q_E	Thermal energy added by lighting and appliance load to air flowing through shelter space in unit time
Q_f	Thermal energy added to air stream in unit time by fan or blower
ΣQ_i	Total rate of heat addition to air flowing through shelter space
q_M	Per capita metabolic rate
q_T	Rate of total metabolic heat loss, per person
q_L	Rate of latent metabolic heat loss, per person
q_S	Rate of sensible metabolic heat loss per person
q_R	Radiation component of q_s
q_C	Convection component of q_s

q_E	Thermal energy added to air in unit time by lighting and appliance load, per person
q_Z	Net thermal energy exchange at a unit area of boundary surface in unit time
q_f	Thermal energy added by fan to air stream, per unit mass of dry air
q_r	Net rate of heat transfer by radiation between an object and enclosing surfaces
$\bar{q}_L$	Enthalpy of water vapor associated with latent metabolic heat loss, per person
$\bar{q}_T$	Total heat added to air in unit time by sensible metabolic heat loss and enthalpy of evaporated sweat, per person
$\bar{R}$	Universal gas constant
R	Gas constant for moist air. Also, the ratio of absolute surface temperatures in radiant heat transfer between an object and enclosure
R_a	Gas constant for dry air
R_w	Gas constant for water vapor
r_a	Ratio of dry-air mass velocities in fresh and total air flow rates
S	Slope coefficient for effective temperature lines in temperature-vapor pressure coordinates. In Equations 68 and 70, $S = \sqrt{p_{ws}}$.
T	Absolute temperature of air. In Equations 89 and 90, T_1 and T_2 are absolute surface temperatures.
t	Dry-bulb temperature of air. In Figure 12, t_1 and t_2 are surface temperatures.

t_d	Dew-point temperature of air
t_e	Effective temperature in an environment
t_ω	Temperature of drinking water
t^*	Thermodynamic wet-bulb temperature
V	Volume of a typical air sample. When an analysis is concerned with rates of fluid flow, the sample represents the volume, $\dot{V}$, of air flowing in unit time.
v	Specific volume of moist air
v_a	Specific volume of dry air component
v_w	Specific volume of water vapor component
$\bar{v}$	Volume of one mass mole of air or vapor
W	Humidity ratio of moist air
dW_L, dW_Z	Differential changes in humidity ratio due respectively to evaporation of sweat and to condensation or evaporation of moisture at boundary surfaces.
ΔW	Incremental change in humidity ratio of moist air
x	Variable distance along length of shelter in direction of air flow
X	Mole fraction of component in gaseous mixture
Y	Cumulative number of persons in shelter space in direction of air flow
Z	Perimeter of shelter cross section perpendicular to direction of air flow
z_i	One of "j" elements of the perimeter, Z, characterized by different thermal parameters
β	Length of shelter space in the direction of air flow

ρ Density of moist air

ρ_a Density of dry air component

ρ_w Density of water vapor component

ϕ Relative humidity of moist air

μ Degree of saturation of moist air

ϵ_o Emissivity of object surface relative to black body

ϵ_e Emissivity of enclosure surface relative to black body.

SUBSCRIPTS

0, 1, 2 ... Identification numbers for stations in system at which quantities are evaluated

a Parameter is associated with dry air component

w Parameter is associated with water vapor

ω Parameter is associated with liquid water

s Parameter is evaluated at the saturated condition and, unless the symbol is further modified, at the same dry-bulb temperature

d Parameter is evaluated at the dew point

e Parameter is evaluated at the effective temperature. In Equation 89, the parameter is associated with surface of enclosure.

o Parameter is associated with oxygen component. In Equation 89, the parameter is associated with surface of an object in an enclosure.

c Parameter is associated with carbon dioxide

I	Parameter is associated with inert gases, nitrogen and argon
i	General reference to one member of a series or group of quantities

SUPERSCRIPTS

*	Parameter is evaluated at the thermodynamic wet-bulb temperature

I INTRODUCTION

Among the factors that characterize a shelter program, there are interrelationships that exert a strong influence on viability of the projected system. During the development of a system and its elements, these constraints should be considered collectively and placed in realistic perspective. With this premise, studies under OCD Work Unit 1236A, Effectiveness of Environmental Control, are concerned with reciprocal effects of shelter structure and control system configurations on overall cost-effectiveness as well as with the group of problems that pertains strictly to the physical environment. Emphasis is directed to shelters that afford protection from primary and secondary weapons effects associated with both short and long duration, peak, freefield overpressures in the 10-20 psi range. Specific objectives in this first phase of the studies are to: (1) develop conceptual models of shelter configurations and electromechanical systems for use in parametric studies of system performance and overall cost-effectiveness, (2) modify and evaluate analytical techniques for application to nonuniform environments and control system performance in cold as well as hot seasons, (3) determine values and derive analytic approximations for parameters, and (4) prepare convenient reference data relating to environmental analysis and control

Background

In the evolution of species, all surviving forms of life have become adapted to the environment in which they must live. Man is a terrestrial species, and the normal chemical composition of atmospheric air at or near sea level provides an optimum environment for metabolic processes that produce the energy essential for his various activities.

He can tolerate substantial reductions in oxygen concentration and, to a lesser degree, the associated increase in carbon dioxide concentration. However, minute concentrations of certain toxic gases such as carbon monoxide may be fatal. Man's mechanism for energy conversion is quite inefficient, and the surplus chemical energy expended appears as thermal energy in his body. This surplus energy tends to increase progressively the temperature of his body. For survival, however, the temperature of his body must be maintained within rather narrow limits; and, for comfort, within narrower limits. At some point within these limits, the nature of man's environment must therefore be favorable to removal of this metabolic heat from his body, that is, favorable to establishment of a state of thermal equilibrium.

Civil defense shelters for protection against various weapons effects must therefore cater to man's inherent biophysical characteristics and provide a probable environment in which body temperatures will remain within safe upper and lower limits. In particular, environmental states near the threshold for thermal equilibrium are most precarious. Here, metabolic heat is transferred largely by evaporation, and time-dependent physiological effects of body dehydration as well as temperature must be considered. Situations accompanied by a general increase in physical activity could become critical within an hour.

In research projects concerned with the development of technology, data, systems, and hardware for control of shelter environments, emphasis has been placed on requirements for fallout shelters that have been identified under the National Fallout Shelter Survey. Although the fundamental principles for environmental control apply generally to all types of shelters, the constraints, assumptions, techniques, and configurations are subject to wide variations. Some of these differences are apparent among the thousands of identified shelter spaces, which are no more alike than diverse parts of the buildings that contain them.

Well over half of these shelter spaces are located on aboveground floors of existing buildings, and natural ventilation appears to be adequate for many of these spaces. For a specific shelter, reasonable doubts can be removed only by a reliable predictive scheme that considers all significant effects. Are window openings relatively large and distributed on both windward and leeward sides? Do interior partitions and doors restrict air flow excessively? Does compartmentation promote air distribution and movement through shelter spaces? Are prevailing winds suitable with respect to velocity and direction? Do neighboring obstructions divert kinetic wind forces?

Identified fallout shelters in basements of existing buildings generally provide better protection for the occupants but pose environmental problems that relate more closely to those in underground blast shelters. If the per capita area of surfaces in contact with relatively cool earth is large, conductive effects may contribute significantly to removal of metabolic heat and stabilization of environmental conditions. However, basements of buildings are often heated. Initial earth temperatures then tend to be higher than corresponding temperatures adjacent to unheated spaces. Since these basement fallout shelters are located in existing public or private buildings, opportunities for making structural modifications or installing fixed equipment are restricted, and portable hardware is appropriate. The need for isolating the environment in these fallout shelters from weapons effects is largely concerned with shielding occupants from radioactive fallout deposited outside the shelters. This essentially justifies the utilization of any and all available openings for intake and exhaust air, and also facilitates the use of pedal-powered ventilating fans with low-pressure, high-volume characteristics.

The degree of isolation required in blast-resistant shelters depends on design assumptions with respect to peak overpressure and duration of the positive pressure phase. For high design overpressures, blast closures at door and ventilation openings are obviously necessary. In shelters intended for low-order protection from primary weapons effects, attenuating devices at ventilating openings may suffice.

In most studies of shelter environments, analyses have been based on the assumption that temperatures and humidities within the space are uniform and correspond to the state of the leaving air. This assumption cannot be made in a study of spatial and transient variations. The uniform or isostate environment represents a special case under the more general assumption of variability. A uniform environment could be approached at any time by (1) using a very high rate of ventilation with either fresh or recirculated air or (2) distributing the air in precise accordance with the distribution of heat and moisture loads. Moreover, most studies have been concerned with resultant effective temperatures in the range of 75° to 90° and with metabolic parameters associated with this range. To determine the variable environmental conditions that would develop during cold as well as hot weather and to evaluate the benefits to be derived from partial recirculation of air, the range of metabolic data must be extended. People in environments that vary through a wide range of temperatures can be assumed to wear clothing that is suitable under the prevailing conditions at any point and time; that is the clothing would be optimum. Revised metabolic parameters should therefore reflect this assumption and should be consistent with both empirical data and the principles of heat-mass transfer. In a hot environment, a minimum of clothing would be optimum; in a cool environment, heavier clothing would be appropriate.

Acknowledgments

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II MODEL SHELTERS AND FIXED EQUIPMENT

General Considerations

A series of model shelter configurations is needed to provide a basis for delineation of environmental control systems, analytical studies of system performance, and derivation of comparative cost data. If the matrix includes three shelter configurations, three shelter sizes, five variant environmental control systems, three climatic locations, and four soil types, a total of 540 shelter-system-ambient combinations must be considered for comparisons of system performance and overall cost-effectiveness. The first type of soil would have a thermal conductivity, K=0; that is, the shelter would be adiabatic. From this matrix, representative combinations can be selected as necessary to determine effects of parametric variations. Typical configurations must be developed in sufficient detail to support meaningful cost estimates.

Model Shelter Configurations

The basic plan shown in Figure 1 is a composite representation for three types of shelters. In each type, there is an identical central bay that contains an assembly room, in which occupants would probably be somewhat active, and utility areas, in which occupancy would be incidental or dependent on circumstances. The shelter concepts differ significantly with respect to the structural treatment of dormitory facilities and, consequently, with respect to the total area of surfaces in contact with adjacent masses of earth. In the first, Type A, there is a rectangular dormitory room along each side of the central bay. In the second, Type B, there is a group of three parallel dormitory galleries along each side

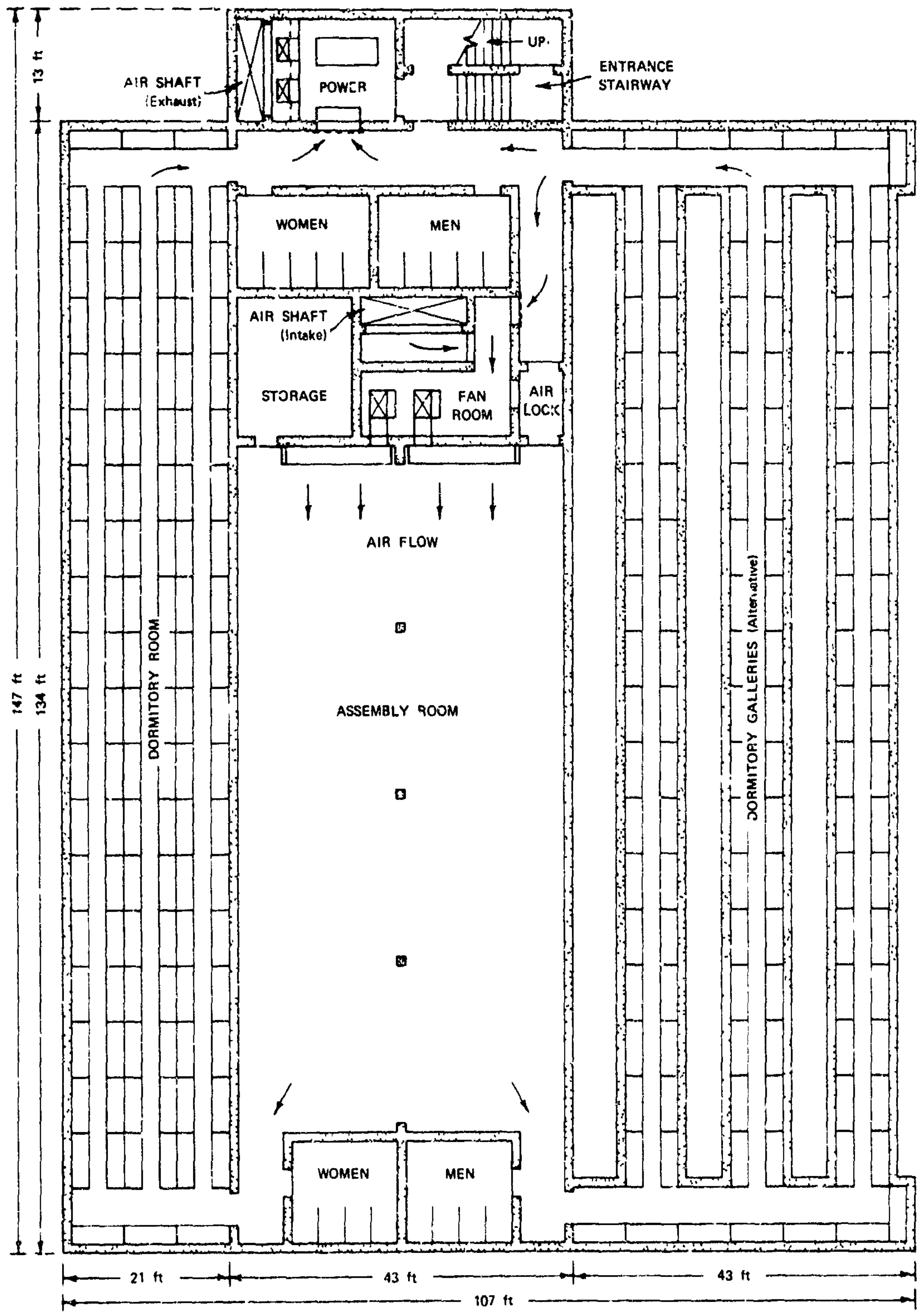

FIGURE 1 BASIC PLAN FOR MODEL SHELTERS (Composite of Rectangular and Gallery Types)

of the central bay. These galleries may be made of either corrugated steel or precast concrete pipe at least 7.5 feet in diameter. Standard pipe sections 8 feet in diameter or equivalent cattle-pass sections could be used to advantage. In the third or composite configuration, Type AB, there is a rectangular dormitory room along one side of the central bay and a group of three parallel dormitory galleries along the opposite side, as shown in Figure 1. A salient feature common to all of these arrangements is that the compartmentation and interconnecting openings provide for distribution of ventilating air from the fan room to all normally occupied spaces with virtually no ductwork. An air lock in the corridor adjacent to the fan room prevents short circuiting of air flow through the shelter. Alternatively, a revolving door could be used for this purpose. If a single door were used, small pressure differentials would interfere with opening and closing, and intermittent traffic would disrupt the normal pattern of air flow.

The capacities and dimensions of these shelters have been synthesized from the floor area required for two bunk spaces separated by an aisle, and this area in turn relates to a linear module of 1.1 feet. Each bunk space and each aisle is 2 modules (2.2. feet) wide and 6 modules (6.6 feet) long. Each side of the square area allocated to the unit group of two bunk spaces separated by one aisle space is therefore 6.6 feet or 2.0 meters long, and the corresponding floor area is 43.56 square feet or 4.0 square meters. On the basis of this unit group of bunk spaces, per capita floor areas would be 21.78, 10.89, 7.26, or 5.44 square feet per person, if bunk heights were, respectively, one, two three, or four decks. For a realistic degree of austerity, single-deck berthing can hardly be justified. However, the value of 7.26 square feet per person associated with triple-deck bunks is consistent with a reasonable degree of austerity in the assembly room. This affords a convenient basis for correspondence or interchangeability between space

allocations in passive and active areas. Plans for the model shelters are based on this criterion for interchangeability, with a further assumption that the numbers of people in passive and active areas are equal; that is, each person would spend half of each day in each of these areas. The capacity of the active or assembly area in all configurations corresponds to its berthing capacity, if it were filled with groups of triple-deck bunks.

The inside width of the rectangular dormitory rooms is 18 linear modules or 19.8 feet, which is the dimension required for three parallel aisles and adjacent bunk spaces. Wall thicknesses are tentatively assumed to be 1 module or 1.1 feet; this may be modified by structural considerations. The clear span for the concrete roof slab is therefore about 20 feet. This width provides for efficient use of space and was selected after consultation with structural engineers. A clear span of 20 feet approaches the maximum span for economical structures designed to resist long-duration peak overpressures in the 15-20 psi range. The length of each row of 18 bunk spaces is 118.8 feet, and three extra bunk spaces are available in the corridor at each end of the room.

In each gallery-type dormitory facility, seven bunk spaces are available in the access corridor at each end of the three pipe galleries, the center lines of which are separated by a distance equal to the length of two bunk spaces or 13.2 feet. Optimum gallery spacing is one of the elements to be investigated in these studies. The space around each gallery is filled with earth. Otherwise the gallery plan is similar to the arrangement in a rectangular dormitory room, except that the bunk height in the pipe galleries is limited to two decks. Greater bunk heights could be used if the galleries were rectangular in cross section with a ceiling height of about 8 feet, but this would considerably reduce the thermal benefits to be derived from use of galleries for increasing

the per capita area of surfaces in contact with adjacent masses of earth. In general, pipe galleries are structurally and economically advantageous for use as blast-resistant shelter.

The dormitory facilities provide return paths for air to be either recirculated through the fan room and shelter or exhausted to atmosphere through the emergency power room, entryway and toilets. Small vent stacks or exhaust ducts are required for the toilets shown in Figure 1. To obtain an approximately uniform velocity of air movement in the Type A shelters, the width of the assembly room should be about twice the width of each rectangular dormitory room. The same air velocity would also be obtained in dormitory galleries, if the pipe sections were 8.2 feet in diameter, but this is not critical. Spans for the room slab in the assembly and dormitory rooms have therefore been equalized, and beam-and-column support is provided along the center line for the two spans in the assembly room. A bearing wall that divides the assembly room into two halves appears to be undesirable from the standpoint of space utilization. Placement of relatively narrow dormitory rooms or galleries on opposite sides of the central bay serves to improve ventilation in corners of both assembly and dormitory spaces. Weighted fabric curtains or brattices can be used to balance rates of air flow along parallel return paths.

Capacities of Model Shelters

The berthing arrangement shown in Figure 1 for the Type AB composite plan is based on double-deck bunks in all dormitory spaces. If the shelter is fully occupied, there are 228 people in the rectangular dormitory room, 244 people in dormitory galleries, and 472 people in the assembly room for a total of 944 persons. The active and passive areas are separated by a partition, and the unit floor area in the assembly room is 6.8 square feet per person.

If bunk height in the rectangular dormitory room is increased to three decks, provision of adequate space for active persons requires that each row of bunks in this room be decreased in length from 18 to 15 bunks. The active area can then expand into space made available by omitted bunks. The shelter can now accomodate 279 people in the dormitory room, 244 people in the dormitory galleries and 523 people in active areas for a total of 1,046 persons. The unit floor area in active areas would be 7.2 square feet per person.

If bunk height in the rectangular dormitory room is again increased to four decks, each row of bunks in this room should be further decreased in length from 15 to 13 bunks, and the active area should again expand. The shelter can now accommodate 324 people in the dormitory room, 244 people in dormitory galleries, and 568 people in active areas for a total of 1136 persons. The unit floor area in active areas would be 7.0 square feet per person.

The effect of increasing bunk height in the Type A configuration, which has two rectangular dormitory rooms, would be similar, except that the increments in shelter capacity would be doubled. In the Type B configuration, which has a set of three parallel pipe galleries on each side of the central bay, such increases in bunk height and shelter capacity are not practicable.

The rectangular dormitory room shown in Figure 1 is not advantageous for use of end-entry bunks; widths in multiples of about 16 feet would be better suited to this purpose. This adaptation would represent an additional shelter configuration. End-entry bunks are not practicable in pipe galleries.

Capacities of three variant types of shelters and associated surface areas are summarized in Table 1. Total floor areas of the Type A, Type B, and Type AB shelters are, respectively, 10,790, 11,370 and 11,080 square

feet. These areas include the entryway as well as utility rooms and corridors in the central bay. The small differences in floor area are caused by the added space in access corridors that connect separated ends of the parallel pipe galleries. Aggregate floor areas of the entryway, utility rooms, and corridors in the central bay constitute 21.1, 20.0, and 20.6 percent of the total floor area, respectively, for the Type A, Type B, and Type AB configurations. If the unit floor areas tabulated in column seven of Table 1 were based on the total floor area of the shelter rather than the sum of floor areas in the assembly room and berthing spaces, the values would be increased by 28, 26, or 27 percent, respectively, in the Type A, Type B, and Type AB configurations. In the last column of Table 1, the effective areas of surface contiguous to earth do not include surfaces in the entryway, power room, storage room, and toilets. The floor areas of available space in the assembly room, one dormitory room, and one set of three galleries with connecting corridors are 3219, 2614, and 2904 square feet respectively. In each dormitory room, the deletion of three or five tiers of bunks from one end of each row increases the available active area by 523 or 784 square feet, respectively, if bunks along the adjacent end wall are also deleted.

TABLE 1

DATA ASSOCIATED WITH MODEL SHELTERS

TYPE OF SHELTER *	HEIGHT OF BUNKS (Decks)		NOMINAL CAPACITIES (Number of People)			UNIT FLOOR AREA ** SqFt/Person	EFFECTIVE SURFACE CONTIGUOUS TO EARTH SqFt/Person
	IN DORMITORY ROOMS	IN DORMITORY GALLERIES	IN BERTHING AREAS	IN ASSEMBLY AREAS	TOTAL IN SHELTER		
A	2	--	456	456	912	9.27	23 41
	3	--	558	558	1116	7 57	19 13
	4	--	648	648	1296	6.52	16 47
B	--	2	488	488	976	9 25	32 02
AB	2	2	472	472	944	9.26	27.86
	3	2	523	523	1046	8.36	25.14
	4	2	569	568	1136	7 69	23 15

* Type A Rectangular dormitory room on each side of central bay
Type B Dormitory galleries on each side of central bay.
Type AB: Composite plan with dormitory room and dormitory galleries.

** Based on total floor area of assembly room and berthing spaces only

Environmental Control Systems

The following types of environmental control systems are suitable for use in conjunction with model shelters that are generally similar to those shown in Figure 1:

System 1--Basic ventilating system that supplies 100 percent air to the shelter, without recirculation.

System 2--Modified ventilating system that supplies a mixture of fresh and recirculated air to the shelter, with an adjustable fresh/total air ratio of 0 to 100 percent.

System 3--Modified ventilating system that supplies a variable mixture of fresh and recirculated air to the shelter, with the added capability for evaporative cooling of fresh air with recirculated air.

System 4--Modified ventilating system that supplies a variable mixture of fresh and recirculated air to the shelter, with the added capability for cooling with well water.

System 5--Modified ventilating system that supplies a variable mixture of fresh and recirculated air to the shelter, with the added capability for cooling and dehumidifying with chilled water.

All of these systems are derived from the basic ventilating system, and this basic system is a special version of System 2. However, the performance characteristics and parameters associated with the systems are quite different; each successive system is more capable than the system from which it is derived. When these systems are used in shelters having the configurations shown in Figure 1, two short ducts are needed for connecting fan discharge openings to diffusion outlets at the fan room end of the assembly room. Small ducts or vent stacks are also needed to exhaust air from toilet rooms. Most of the air is exhausted through the power room and entryway. Two double-inlet, double-width centrifugal fans with a total capacity of about 15,000 cfm are shown. These fans could be moved through a 36-inch door opening.

Systems 3, 4, and 5 use apparatus for heat exchange between fresh air and water. The concept shown in Figure 2 includes a rock grille in the fresh air intake shaft. This rock grille is wet by water sprays and serves as a heat exchanger for any or all of these systems. A suitable material for these rock grilles would be approximately spherical cobbles from two to four inches in diameter. This rock grille:

1. Attenuates short-duration blast overpressures.
2. Attenuates thermal pulses of hot air.
3. Reduces amplitude of diurnal temperature cycle for fresh air supply.
4. Cools air adiabatically by evaporation when wet with recirculated water.
5. Cools and dehumidifies air by direct contact when wet with chilled water or relatively cool well water.
6. Removes particulate matter and soluble gases by washing or scrubbing action.

In addition to the array of spray nozzles, spray header, piping and water basin, a circulating pump is necessary. For higher design overpressures, a blast-actuated closure would also be needed in the ventilation shaft, as indicated in Figure 2.

Since commercial power may not be available, an emergency power supply is required for all of these systems. System 5 also includes a package water chiller. Both of these auxiliary items must reject heat to some available sink. The vent shaft for air exhausted through the power room could serve as the heat sink for either the engine-generator or the water chiller, if this vent shaft incorporated a wet rock grille similar to that shown in Figure 2. This wet rock grille would serve as a packed cooling tower for cooling water by evaporation. Waste heat from an engine-generator should be considered for use in tempering or heating cold air during the winter season.

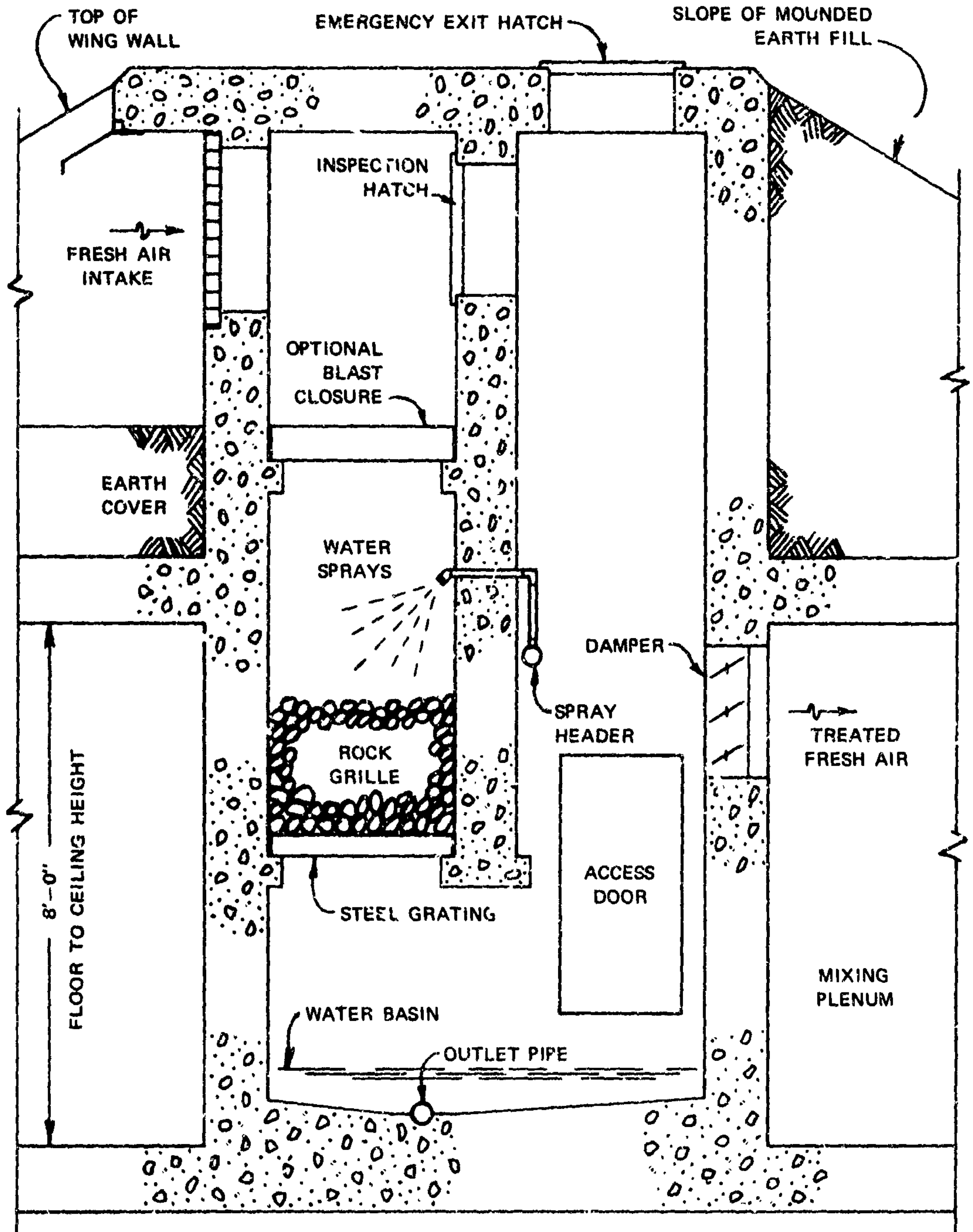

FIGURE 2 CONCEPT OF AIR INTAKE FACILITY FOR BLAST-RESISTANT SHELTERS

If these shelters were used only for fallout protection, the rock grilles and blast closures in vent shafts for fresh and exhaust air could be omitted. The water sprays and associated apparatus would be retained. These omissions would reduce the system resistance to air flow and would facilitate the expedient use of pedal-powered fans or punkahs. If the fan room is large enough to accommodate the required number of these devices, they could be connected to the fan discharge ducts shown in Figure 1.

In Table 1, the highest value for the effective area of surfaces contiguous to earth is about 32 square feet per person in the Type B shelter. It is anticipated that heat conduction effects associated with this area will assist in shelter cooling, but will not be sufficient under probable conditions to provide all cooling requirements during an occupancy period of two weeks. At an increase in cost, these surfaces could be increased by doubling the number of galleries and using single-deck bunks. Alternatively, two or three small, self-contained air conditioning units could be installed in the shelter and operated before an emergency. This would reduce the temperature of earth adjacent to the shelter and increase the contribution of earth conduction in cooling the shelter during an emergency.

III ENVIRONMENTAL ANALYSIS

Ventilation with Mixed Air

The effectiveness of an environmental control system depends on capacities and configurations of the system and the shelter, ambient conditions with respect to weather and contiguous earth, and characteristics of internal heat and moisture loads. For a generalized analysis that considers spatial, seasonal, and transient variations, the associated variations in significant parameters must be evaluated.

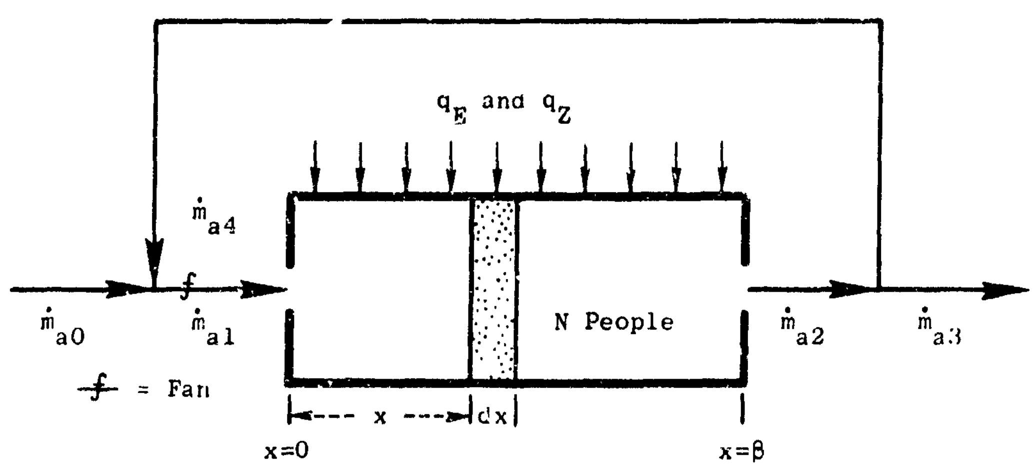

FIGURE 3 VENTILATION OF SHELTER SPACE WITH FRESH AND RECIRCULATED AIR

The basic elements of a shelter with a ventilating system that provides for partial recirculation of air are shown diagrammatically in Figure 3. Central air moving and conditioning apparatus can be added to this system without changing the fundamental analysis of processes that take place within occupied spaces. With this configuration, there are two interrelated processes in the analysis: (1) the mixing of fresh and recirculated air and (2) the changes in temperature, humidity, and enthalpy

of air as it passes through the shelter. In these processes, energy and mass are conserved. The quantities, $\dot{m}_{a0}$, $\dot{m}_{a1}$, $\dot{m}_{a2}$, $\dot{m}_{a3}$, and $\dot{m}_{a4}$, are mass flow rates of dry air at observation stations 0, 1, 2, 3, and 4 (pounds of dry air per hour). In general, the air at all these stations is moist air with humidity ratios of W_0, W_1, W_2, W_3, and W_4 pounds of moisture per pound of dry air, and specific enthalpies of h_0, h_1, h_2, h_3, and h_4 Btu per pound of dry air. Then, the corresponding mass flow rates for water vapor are $\dot{m}_{w0} = \dot{m}_{a0}W_0$, $\dot{m}_{w1} = \dot{m}_{a1}W_1$, $\dot{m}_{w2} = \dot{m}_{a2}W_2$, $\dot{m}_{w3} = \dot{m}_{a3}W_3$, and $\dot{m}_{w4} = \dot{m}_{a4}W_4$ in pounds of water vapor per hour. Similarly, the total enthalpies are related to specific enthalpies at any station, i, by the equations, $H_i = \dot{m}_{ai}h_i$, in Btu per hour. Thermal energy added to the air by fan losses is Q_1 $\dot{m}_{a1}q_f$ Btu per hour. Since the humidity ratio and specific enthalpy are properties that determine a psychrometric state, $W_4 = W_2$ and $h_4 = h_2$. Also, since oxidation processes in the shelter are assumed to have negligible effect, the dry air component is essentially invariant, and $\dot{m}_{a2} = \dot{m}_{a1}$.

For the mixing process, the following relationships apply:

$$\dot{m}_{a1} = \dot{m}_{a0} + \dot{m}_{a4}$$

$$\dot{m}_{w1} = \dot{m}_{w0} + \dot{m}_{w4}$$

$$H_1 = H_0 + H_4 + Q_f$$

$$\dot{m}_{a1}h_1 = \dot{m}_{a0}h_0 + \dot{m}_{a4}h_2 + \dot{m}_{a1}q_f$$

$$\dot{m}_{a1}W_1 = \dot{m}_{a0}W_0 + \dot{m}_{a4}W_2. \tag{1}$$

Then,

$$h_1 = \frac{\dot{m}_{a0}h_0 + \dot{m}_{a4}h_2}{\dot{m}_{a1}} + q_f \tag{2}$$

$$W_1 = \frac{\dot{m}_{a0} W_0 + \dot{m}_{a4} W_2}{\dot{m}_{a1}} \qquad (3)$$

If the mass ratio of fresh to total air flow is $r_a = \frac{\dot{m}_{a0}}{\dot{m}_{a1}}$ on a dry air basis, Equations 2 and 3 become

$$h_1 = h_2 - r_a (h_2 - h_0) + q_f \qquad (4)$$

and

$$W_1 = W_2 - r_a (W_2 - W_0). \qquad (5)$$

These equations determine the psychrometric state of the fresh-recirculated air mixture supplied to the occupied space.

In the shelter, heat and moisture are added to the flowing air. For present purposes, the net totals of heat and moisture added by all sources in the occupied space can be represented by the symbols ΣQ_Δ for heat and $\Sigma \bar{m}_{w\Delta}$ for moisture. Then, since $\dot{m}_{a2} = \dot{m}_{a1}$,

$$\Sigma Q_\Delta = \dot{m}_{a2} h_2 - \dot{m}_{a1} h_1 = \dot{m}_{a1} (h_2 - h_1)$$

and

$$\Sigma \bar{m}_{w\Delta} = \dot{m}_{a2} W_2 - \dot{m}_{a1} W_1 = \dot{m}_{a1} (W_2 - W_1). \qquad (6)$$

Therefore, the change in specific enthalpy is

$$\Delta h = h_2 - h_1 = \frac{1}{\dot{m}_{a1}} \Sigma Q_\Delta \qquad (7)$$

the change in humidity ratio is

$$\Delta W = W_2 - W_1 = \frac{1}{\dot{m}_{a1}} \Sigma \bar{m}_{w\Delta} \qquad (8)$$

and the <u>average</u> slope of the process line in the space is

$$\frac{\Delta h}{\Delta W} = \frac{h_2 - h_1}{W_2 - W_1} = \frac{\Sigma Q_\Delta}{\Sigma \bar{m}_{w\Delta}} \qquad (9)$$

where Δh and ΔW are incremental changes in specific enthalpy and humidity ratio.[1*]

The specific enthalpy and humidity ratio of air leaving the shelter space can be found by combining Equations 4 and 7 and Equations 5 and 8. Then, since $r_a = \dot{m}_{a0}/\dot{m}_{a1}$, the specific enthalpy is

$$h_2 = h_3 = h_4 = h_0 + \frac{1}{\dot{m}_{a0}} \left[Q_f + \Sigma Q_\Delta\right] \tag{10}$$

and the humidity ratio is

$$W_2 = W_3 = W_4 = W_0 + \frac{1}{\dot{m}_{a0}} \Sigma \bar{\dot{m}}_{w\Delta} . \tag{11}$$

These equations determine the psychrometric state of moist air leaving the shelter.

If the shelter environment were uniform, the slope of the process line would be constant and equal to the average slope determined in Equation 9. The state of this uniform environment would then be determined by the amount and condition of outside air provided. However, such a uniform state can be approached only by using ventilating rates that are large relative to heat and moisture loads or by distributing the air substantially in accordance with the distribution of loads in the shelter. In general, the environment will not be uniform and the slope of the process line will not be constant, because the heat and moisture sources are sensitive to temperature and humidity.[2] Spatial variations in the thermal environment are likely to be most significant when associated with cold weather, shelters with large or compartmented spaces, and ventilating systems that neither provide for recirculation of return air nor prevent random infiltration of outside air.

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* Numerals correspond to items listed in the References section near the end of the report

Heat and Moisture Loads in Shelters

Shelter heat and moisture loads are assumed to include all effects that tend either to increase or to decrease the temperature, enthalpy, or humidity of air traversing the shelter space. In general, these effects include metabolic heat and moisture from the occupants, heat added by lights and appliances, heat transmitted through boundary surfaces to the ambient atmosphere, and heat exchanged by conductive effects in contiguous masses of earth or construction materials. Moisture may be removed from the air by condensation, if boundary surface temperatures are below the dew point, or may be added to the air by evaporation, if surfaces are damp and relatively warm. Equipment such as engine generators must be cooled, but heat emitted by this apparatus will not affect the environment in occupied spaces, if the apparatus is located in a separate room and is cooled by water, a remote radiator, or exhaust air.

The model configuration shown in Figure 3 represents a ventilated shelter with heat and moisture loads caused by occupancy, lighting, and heat flux across boundary surfaces. For analytical purposes, psychrometric conditions are assumed to be uniform over a transverse section of the shelter. Heat or moisture exchanges are considered positive if they tend to increase the corresponding enthalpy or humidity ratio of flowing air.

Metabolic Loads

The shelter space is assumed to be occupied by N people, each of whom is in thermal equilibrium with a metabolic rate of q_M Btu per hour per person. Then, an approximate relationship among heat quantities is

$$q_M = q_T = q_R + q_C + q_L = q_S + q_L \qquad (12)$$

where

q_T = total metabolic heat transferred to the environment (Btu/hour per person),

q_L = latent heat transferred by evaporation of moisture from skin surfaces and respiratory tract,

q_R = metabolic heat transferred by radiation from skin or cloth surfaces,

q_C = metabolic heat transferred by convection from skin or clothing surfaces and from the respiratory tract, and

q_S = $q_R + q_C$ = sensible heat transferred from body to environment.

Energy expenditure or metabolic rate, q_M, depends directly on the level of physical activity in the shelter or indirectly on heat needed to maintain a normal body temperature. A typical value for sedentary male adults is 400 Btu per hour per person, or 20 Btu per hour per square foot of skin surface. For thermal equilibrium, heat must be transferred to the environment at an equivalent rate. In a hot environment, a rise in body temperature is inevitable if either vapor pressure gradients or sweat rates are not sufficient to allow the necessary evaporative cooling on moist skin surfaces. In a neutral or cool environment, total metabolic heat losses associated with a normal body temperature are substantially constant, if the insulating effect of clothing worn at any time is optimum for the prevailing temperature.[3]

Latent heat loss from the body is related to moisture gain by the air. Then, with the premise that water is an incompressible fluid having a specific heat of unity,

$$q_L = \bar{m}_{wL}\left[h_g - 1.0\left(t_\omega - 32\right)\right] \approx \bar{m}_{wL}\, h_{fg} \qquad (13)$$

or

$$\bar{m}_{wL} \approx q_L/h_{fg} \tag{14}$$

where

$\bar{m}_{wL}$ = rate of sweat evaporation from body surfaces (lbm/hour per person),

h_{fg} = latent heat of vaporization for water at skin temperature (Btu/lbm),

h_g = specific enthalpy of saturated water vapor at skin temperature, and

t_ω = temperature of drinking water (^oF).

Tabulated values for enthalpy of saturated water vapor are referred to a 32°F base and include enthalpy of liquid as well as latent heat of vaporization. However, only the latent heat of vaporization is considered effective in removing metabolic heat from the body. Therefore, the enthalpy of water vapor added to the air is

$$\bar{q}_L = q_L + \bar{m}_{wL} h_f = \bar{m}_{wL}(h_f + h_{fg}) = \bar{m}_{wL} h_g \tag{15}$$

where

h_f = enthalpy of liquid water at skin temperature (Btu/lbm).

The total metabolic heat added to the air should also include the heat of liquid. Then

$$\bar{q}_T = q_S + q_L + \bar{m}_{wL} h_f = q_T + \bar{m}_{wL} h_f. \tag{16}$$

For thermal equilibrium with a given activity level, total metabolic heat losses q_T, are essentially constant, but the associated sensible and latent heat fractions are functions of temperature. In general terms, latent heat losses are:

$$q_L = F(t) \text{ Btu per hour per person.} \quad (17)$$

The rate of moisture exchange is

$$\bar{m}_{wL} = F(t)/h_{fg} \text{ lbm per hour per person.} \quad (18)$$

The total heat added to the air is

$$\bar{q}_T = q_T + F(t) \frac{h_f}{h_{fg}} \text{ Btu per hour per person.} \quad (19)$$

The number of people in a differential length of shelter is

$$dY = \frac{N}{\beta} dx \quad (20)$$

where

Y = number of people in shelter from $x = 0$ to $x = x$

β = length of the shelter in direction of air movement

N = total number of persons in shelter space.

The rates of change in specific enthalpy and humidity ratio caused by metabolic loads are

$$dh_M = \frac{\bar{q}_T}{\dot{m}_{a1}} dY = \frac{N}{\beta \dot{m}_{a1}} \bar{q}_T \, dx \quad (21)$$

and

$$dW_L = \frac{\bar{m}_{wL}}{\dot{m}_{a1}} dY = \frac{N \bar{m}_{wL}}{\beta \dot{m}_{a1}} dx \quad (22)$$

in which $\bar{q}_T$ and $\bar{m}_{wL}$ are functions of environmental temperature, as defined in Equations 17, 18, and 19. A rational expression for $q_L = F(t)$ remains to be determined for a wide temperature range.

Constant Heat Loads.

Sensible heat emitted by electric lights and appliances can be considered as a constant and distributed source of heat in a shelter. If the unit wattage of electrical fixtures in the shelter is P_E watts per square foot of floor area, the aggregate sensible heat load is

$$Q_E = 3.413\, P_E A_F \tag{23}$$

where

Q_E = sensible heat load in shelter due to lights (Btu/hour)

P_E = unit wattage of lights (watts/sq ft of floor area)

A_F = floor area of shelter space (sq ft)

The rate of change in specific enthalpy of the air due to lighting is

$$dh_E = \frac{Q_E}{N\dot{m}_{a1}}\, dY = \frac{Q_E}{\beta\dot{m}_{a1}}\, dx \tag{24}$$

Variable Loads

Variable heat and moisture loads in this context are associated with heat flux at boundary surfaces, walls, floor, and ceiling. Heat transfer through these surfaces is always in a state of quasi-equilibrium; that is, the net rate of heat transfer between the environment and the surfaces at any time is equal in magnitude to the rate of heat transfer by conduction at surfaces of contiguous materials. Surface temperatures at any time are determined by this dynamic state of thermal equilibrium.

When air passes through an occupied shelter that is virtually "adiabatic," either because the structure is well insulated or because temperature gradients in contiguous materials are negligible, properties of the air are changed by heat and moisture from interior sources. Moisture is added by evaporation from skin surfaces and respiratory tract.

Heat is added directly to the air by convection. Heat is radiated from lights and from skin or clothing to boundary surfaces and is subsequently added to the air by convection. Even in perfectly insulated shelters, boundary surface temperatures tend to be slightly higher than environmental air temperatures. Heat and moisture are removed from the shelter only by ventilating air.

In general, however, the effects of heat transmission through outside walls and heat conduction in contiguous masses of earth are not negligible. These transmission and conduction effects are likely to be substantial in cold weather, or at any time in shelters that have relatively large areas of surface in contact with earth. In particular, transient effects of heat conduction attenuate both hot and cold extremes in environmental temperature. Moisture may condense on relatively cool interior surfaces and, if construction materials are absorptive, may evaporate as the surfaces gradually increase in temperature.[4,5]

If the transverse perimeter of a shelter is divided into "j" lengths with dissimilar thermal parameters, the total perimeter is

$$Z = \sum_{i=1}^{j} z_i \qquad (25)$$

the total surface area is

$$A_Z = \beta Z, \qquad (26)$$

and the differential area of each discrete part of the surface is

$$dA_i = z_i dx = \frac{\beta}{N} z_i dY. \qquad (27)$$

Each of these differential surface areas is a potential heat and moisture load the magnitude of which depends on temperature differences, vapor pressure gradients, and values of film conductance and vapor diffusion

coefficients. Pending further study, heat and moisture exchanges at boundary surfaces will be indicated symbolically. For energy exchanges,

$$dq_Z = \Sigma\, G_i(\Delta t,\ \Delta p_w) z_i\ dx = \frac{\beta}{N} \Sigma\, G_i(\Delta t,\ \Delta p_w)\ z_i\ dY. \tag{28}$$

For moisture exchanges

$$d\bar{m}_{wZ} = \Sigma\, g_i(\Delta p_w)\ z_i\ dx = \frac{\beta}{N} \Sigma\, g_i(\Delta p_w) z_i\ dY. \tag{29}$$

For changes in specific enthalpy

$$dh_Z = \frac{1}{\dot{m}_{a1}} dq_Z\ . \tag{30}$$

For changes in humidity ratio

$$dW_Z = \frac{1}{\dot{m}_{a1}} d\bar{m}_{wZ}\ . \tag{31}$$

In the above equations, the symbol, $G_i\ (\Delta t,\ \Delta p_w)$, represents a series of rational functions that determines energy transfer rates per unit of surface area in terms of temperature and vapor pressure differences. The symbol, $g_i\ (\Delta_{pw})$, represents a series of rational functions that determines moisture transfer rates per unit of surface area in terms of vapor pressure differences.

Combined Loads

The rate of change in specific enthalpy of air caused by aggregate effects of metabolic, lighting, and conductive heat exchanges is

$$\beta \dot{m}_{a1} \frac{dh}{dx} = N\dot{m}_{a1} \frac{dh}{dY} = Nq_T + \frac{h_f}{h_{fg}}\ NF(t) + Q_E + \beta\, \Sigma\, G_i(\Delta t,\ \Delta p_w)\ z_i. \tag{32}$$

The corresponding rate of change in humidity ratio of air is

$$\beta \dot{m}_{a1} \frac{dW}{dx} = N \dot{m}_{a1} \frac{dW}{dY} = \frac{N}{h_{fg}} F(t) + \beta \Sigma g_i (\Delta p_w) z_i . \tag{33}$$

For an insulated or adiabatic shelter, the last term in each of these two equations vanishes, and the slope of the process line is

$$\frac{dh}{dW} = \frac{q_T + \frac{h_f}{h_{fg}} F(t) + \frac{Q_E}{N}}{\frac{F(t)}{h_{fg}}} . \tag{34}$$

In Equation 34, the slope is a function of shelter capacity, N, as well as environmental temperature. However, if the lighting load, Q_E, were based on Btu per person rather than Btu per square foot of floor area, the slope would become independent of shelter capacity. That is, if $Q_E = Nq_E$

$$\frac{dh}{dW} = \frac{q_T + \frac{h_f}{h_{fg}} F(t) + q_E}{\frac{F(t)}{h_{fg}}} \tag{35}$$

in which q_E is the lighting load in Btu per person. Metabolic equations for latent heat, $q_L = F(t)$, will be considered in a later section.

For the general case in which variable heat and moisture loads must be considered, there are two simultaneous effects, removal of heat and moisture by ventilating air and removal of heat by transient heat conduction in contiguous materials, with or without moisture condensation. These two problems are mutually related by the parameters, temperature of boundary surfaces and pressure of saturated water vapor at the corresponding dew point. Since no analytical solution for the general case has been found, it is practically necessary to obtain numerical solutions by computer simulation of the problems.[6,7,8]

Properties of Moist Air

One of the parameters needed to determine the physical properties or changes in properties of moist air is barometric pressure. Since shelters may be located at altitudes from sea level to about 10,000 feet, a generalized method for environmental analyses should consider barometric pressure as a parameter. At several stages in the derivation of such a method, there are options that affect resultant accuracy. The most precise approach would be based on semiempirical virial equations of state that consider mixing interactions as well as real gas deviations of the constituents from ideal gas relationships. For a barometric pressure of one atmosphere, the thermodynamic properties of moist air at saturation have been determined on this basis and tabulated.[9] The most convenient approach would be based on assumptions that ideal gas relationships and Dalton's law of partial pressures are adequate.[10,11] In the derivation below, ideal gas relationships and Dalton's law are used in conjunction with tabulated values for the real gas properties of water at saturation.[12] Deviations associated with these assumptions will subsequently be examined.

During most air-conditioning processes, there is little or no change in composition of the dry air fraction, and it is convenient to consider moist air as a uniform binary mixture of dry air and water vapor, each of which occupies the same volume of space, simultaneously and independently. Moreover, the composition of the dry air fraction in atmospheric air near ground level does not vary significantly with time or locality and is substantially as shown in Table 2.[13] The apparent molecular weight, M_a, of dry air with the tabulated composition is 28.966 units of mass per mass mole. The effects of variations in carbon dioxide and water vapor concentrations on apparent molecular weight of the mixture will be considered later.

Table 2

COMPOSITION OF DRY AIR NEAR GROUND LEVEL

Constituent Gas	Molecular Weight	Composition (percent) Volumetric	Gravimetric
Nitrogen	28.016	78.090%	75.528%
Argon	39.944	0.930	1.282
Oxygen	32.000	20.950	23.144
Carbon dioxide	44.011	0.030	0.046
		100.000%	100.000%

For a mixture such as moist air, the perfect gas equation may be written in the following alternative forms:

$$pV = pmv = n\bar{R}T = \frac{m\bar{R}T}{M} = mRT \qquad (36)$$

in which

p = total or barometric pressure (lbf)/(sq ft)

V = mv = volume (cu ft)

v = specific volume (cu ft)/(lbm)

m = mass of mixture (lbm)

n = m/M = number of moles of mixture (lb moles)

M = molecular weight of mixture (lbm)/(lb/mole)

T $t + 459.67$ = absolute temperature (^{o}R)

t = temperature (^{o}F)

$\bar{R}$ = universal gas constant = 1545.33 (lbf)(ft)/(lb mole)(^{o}R)

R = gas constant for mixture (lbf)(ft)/(lbm)(^{o}R)

Similar equations can be written for the dry air and water vapor:

$$p_aV = p_a m_a v_a = n_a\bar{R}T = \frac{m_a\bar{R}T}{M_a} = m_a R_a T \tag{37}$$

$$p_wV = p_w m_w v_w = n_w\bar{R}T = \frac{m_w\bar{R}T}{M_w} = m_w R_w T \tag{38}$$

in which parameters are as defined for Equation 36 and the subscripts, a, and, w, refer, respectively, to the dry air and water vapor components.

In accordance with Dalton's law, Equations 36, 37, and 38 are related as follows:

$$p = p_a + p_w. \tag{39}$$

By definition, the humidity ratio, W, is

$$W = \frac{m_w}{m_a} \tag{40}$$

pounds of water vapor per pound of dry air.

Since each of the constituent gases occupy the same volume as the mixture, several equivalent expressions for volume are as follows:

$$V = mv = \frac{n\bar{R}T}{p} = \frac{m\bar{R}T}{pM} = \frac{mRT}{p} = m_a v_a = \frac{n_a\bar{R}T}{p_a} = \frac{m_a\bar{R}T}{p_a M_a} = \frac{m_a R_a T}{p_a}$$

$$= m_w v_w = \frac{n_w\bar{R}T}{p_w} = \frac{m_w\bar{R}T}{p_w M_w} = \frac{m_w R_w T}{p_w}. \tag{41}$$

From Equations 36 through 39,

$$p = \frac{m\bar{R}T}{MV} = \frac{m_a\bar{R}T}{M_a V} + \frac{m_w\bar{R}T}{M_w V}$$

and

$$\frac{m}{M} = \frac{m_a + m_w}{M} = \frac{m_a}{M_a} + \frac{m_w}{M_w} .$$

Then, since $W = \dfrac{m_w}{m_a}$, the apparent molecular weight of the moist air mixture is

$$M = M_w \frac{W + 1}{W + \dfrac{M_w}{M_a}} = 18.016 \frac{W + 1}{W + 0.62197} . \tag{42}$$

The partial densities of dry air and water vapor are also additive. Then

$$\lambda = \lambda_a + \lambda_w \tag{43}$$

where

$\lambda = 1/v =$ density of mixture (lbm)/(cu ft),

$\lambda_a = 1/v_a =$ density of dry air, and

$\lambda_w = 1/v_w =$ density of water vapor.

From the volumetric identities in Equation 41

$$\frac{m_w v_w}{m_a v_a} = \frac{n_w p_a}{n_a p_w} = \frac{m_w p_a M_a}{m_a p_w M_w} = \frac{m_w p_a R_w}{m_a p_w R_a} = 1 . \tag{44}$$

Then, several equivalent expressions for humidity ratio are

$$W = \frac{m_w}{m_a} = \frac{v_a}{v_w} = \frac{\lambda_w}{\lambda_a} = \frac{p_w M_w}{p_a M_a} = \frac{p_w}{p - p_w}\left(\frac{M_w}{M_a}\right) . \tag{45}$$

From the last of these equivalents

$$p_w = \frac{pW}{W + \frac{M_w}{M_a}} = \frac{pW}{W + 0.62197} . \qquad (46)$$

In terms of humidity ratio, the unit mass of a moist air mixture is

$$\frac{m}{m_a} = \frac{m_a + m_w}{m_a} = 1 + W \qquad (47)$$

pounds of moist air per pound of dry air. In terms of humidity ratio, the specific volume of dry air, water vapor, or moist air mixture, <u>referred to dry air</u>, is

$$v_a = \frac{V}{m_a} = \frac{\bar{R}T}{p_a M_a} = W \frac{\bar{R}T}{p_w M_w} = \frac{R_a T}{p}\left(1 + \frac{M_a}{M_w} W\right)$$
$$= \frac{R_a T}{p}(1 + 1.60779W) = \frac{R_w T}{p}\left(W + \frac{M_w}{M_a}\right) = \frac{R_w T}{p}(W + 0.62197) \qquad (48)$$

cubic feet per pound of dry air.

Values of the gas constants for dry air and water vapor are

$$R_a = 53.35 \text{ and } R_w = 85.78.$$

The degree of saturation, μ, is

$$\mu = \frac{W}{W_s} = \emptyset \frac{p - p_{ws}}{p - \emptyset p_{ws}} \qquad (49)$$

and the relative humidity, $\emptyset$, is

$$\emptyset = \frac{p_w}{p_{ws}} = \frac{\mu p}{p-(1-\mu)p_{ws}} . \qquad (50)$$

The subscript, s, indicates that the corresponding parameter is evaluated for the saturated state at the same temperature.

Equation 46 provides a means for determining the partial pressure of water vapor when the humidity ratio is known. An alternative expression that refers to real gas properties of water vapor at saturation is

$$p_w = p_{ws} \frac{W\left(W_s + \frac{M_w}{M_a}\right)}{W_s\left(W + \frac{M_w}{M_a}\right)} . \tag{51}$$

The specific enthalpy of moist air can be determined with reference to either of two points on the saturation line, one of which corresponds to the dry-bulb temperature and the other to the dew point of superheated water vapor in moist air. The equations are, for one pound of dry air:

$$h = h_a + Wh_g = c_{pa}t + Wh_g \tag{52}$$

and

$$h = c_{pa}t + Wh_{gd} + c_{pw}W(t-t_d) \tag{53}$$

where

t = dry-bulb temperature (^oF)

t_d = dew-point temperature (^oF)

h = specific enthalpy of moist air (Btu)/(lb of dry air),

h_a = specific enthalpy of dry air (Btu/(lb of dry air),

W = humidity ratio (lb of vapor)/(lb of dry air),

h_g = enthalpy of saturated water vapor at the mixture temperature, t, (Btu)/(lb of vapor),

h_{gd} = enthalpy of saturated water vapor at the mixture dew point, t_d, (Btu)/(lb of vapor), and

c_{pa} and c_{pw} = specific heats of dry air and water vapor at constant pressure (Btu)/(lbm)(Deg°F).

By equating these two expressions, the condition for equivalence is found to be

$$c_{pw} = \frac{h_g - h_{gd}}{t - t_d} . \tag{54}$$

Values of h_g and h_{gd} can be approximated by a first or second degree polynomial having the general form

$$h_g = A + Bt + Ct^2 \tag{55}$$

$$h_{gd} = A + Bt_d + Ct_d^{\,2}. \tag{56}$$

Then, if Equations 52 and 53 are equivalent, the apparent specific heat of water vapor must be

$$c_{pw} = B + C(t + t_d). \tag{57}$$

Thus, the coefficients of the polynomial expression for h_g determine the apparent specific heat of water vapor. For a first degree polynomial, $C = 0$ and $c_{pw} = B$, a constant.

Equations 52 and 53 are consistent with the ideal gas assumption. However, Equation 52 is preferred because it does not involve the extra parameter, dew-point temperature.

Therefore, Equations 52 and 55 can be combined to interrelate the basic parameters, dry-bulb temperature, humidity ratio, and specific enthalpy

$$h = c_{pa}t + W(A + Bt + Ct^2) = AW + (c_{pa} + BW)t + CWt^2 \tag{58}$$

or

$$CWt^2 + (c_{pa} + BW)t + (AW - h) = 0. \tag{59}$$

This quadratic equation determines the dry-bulb temperature, if values for h and W are known. If the polynomial expression for h_g is of the first degree, C = 0, and the dry-bulb temperature is

$$t = \frac{h - AW}{c_{pa} + BW} . \tag{60}$$

If the polynomial is of second degree, $C \neq 0$, and the quadratic equation must be solved to evaluate the temperature.

The slope of a process line, in terms of specific enthalpy and humidity ratio, is determined by differentiating Equation 58. Then

$$\frac{dh}{dW} = A + Bt + Ct^2 + (c_{pa} + BW)\frac{dt}{dW} + 2CWt\frac{dt}{dW} \tag{61}$$

or, if the polynomial expression for enthalpy of saturated water vapor is of the first degree, C = 0, and the slope is

$$\frac{dh}{dW} = A + Bt + (c_{pa} + BW)\frac{dt}{dW} . \tag{62}$$

This slope is associated with the characteristics of heat and moisture loads in the space, as stated by Equations 9 and 34. Equations 35 and 62 can be combined to form a first order differential equation for the process line in terms of the separable variables, t and W, and containing the latent heat function, $q_L = F(t)$.

Polynomial Approximations

The properties of water vapor at saturation are fundamental to determining the properties of moist air and consequently in the analysis of psychrometric processes. It is convenient to define the essential properties as functional relationships. The specific enthalpy of moist air at any condition involves the enthalpy, h_g, of saturated water vapor at the dry bulb or dew-point temperature of the mixture. Relative humidity, degree of saturation, and humidity ratio at saturation are associated with the pressure of saturated water vapor at the dry bulb temperature of the mixture. The dew-point temperature is determined by the partial pressure of water vapor in the mixture. Tabulated data on the properties of water at saturation have therefore been used to derive polynomial approximations for three parametric relationships by means of a least squares computer program.

These relationships are as follows:

Enthalpy of saturated water vapor as a function of temperature, $h_g = f(t)$.

Pressure of saturated vapor as a function of temperature, $p_{ws} = f(t)$.

Temperature as a function of saturated vapor pressure, $t = f(p_{ws})$.

Since the ranges of temperature extend from -40°F to 160°F, a data source that covers the entire range was selected for this purpose.[12]

Values for enthalpy of saturated water vapor, h_g, and enthalpy of vaporization, h_{fg}, are shown graphically in Figure 4. These enthalpies and the enthalpy of saturated liquid, h_f, are related by the simple equation

$$h_g = h_f + h_{fg} \quad (\mathrm{Btu})/(\mathrm{lbm}). \tag{63}$$

Since $h_f \approx (t-32)$, all of these enthalpies are approximately linear with respect to temperature, and values of h_g will be determined by an equation, $h_g = f(t)$. Polynomial approximations of the first, second, and third degree have therefore been derived for this purpose. Numerical values for the coefficients are given in Table 3, together with the basic data and corresponding values for enthalpy calculated with the polynomial equations. The first degree approximation

$$h_g = 1061.47 + 0.431238t \tag{64}$$

is most convenient and is probably adequate for most applications.

A curve for pressure of saturated water vapor as a function of temperature is shown in Figure 5, together with curves for vapor density, specific volume, and compressibility. The pressure-temperature relationship is obviously far from linear. Data used for deriving polynomial approximations for $p_{ws} = f(t)$ and $t = f(p_{ws})$ are shown in Table 4. Since there is a discontinuity in slope at the ice point, it is advantageous to derive separate equations for ranges above and below 32°F. Also, for equivalent levels of accuracy, the degree of the polynomial can be reduced by using natural logarithms or square roots of vapor pressures rather than tabulated values. This indirect approach is particularly effective for equations that define temperature as a function of vapor pressure.

Within the indicated range of application, all of the following six equations conform closely to the data shown in Table 4 and have an index of determination equal to unity.

Table 3

POLYNOMIAL APPROXIMATIONS FOR ENTHALPY
OF SATURATED WATER VAPOR

Polynomial Form:		$h_g = A + Bt + Ct^2 + Dt^3$		
Degree of Polynomial		1	2	3
Index of Determination		0.999867	0.999994	1.000000
Coefficients:	A	1061.47	1061.00	1061.10
	B	0.431238	0.44683	0.439597
	C	0	-9.80401×10^{-5}	1.53789×10^{-5}
	D	0	0	-4.71574×10^{-7}
TEMPERATURE t (F)	ENTHALPY h_g (Data)	CALCULATED VALUES OF ENTHALPY (Btu)/(Lbm)		
0	1061.09	1061.47	1061.00	1061.10
20	1069.90	1070.10	1069.90	1069.89
32	1075.16	1075.27	1075.20	1075.16
40	1078.68	1078.72	1078.71	1078.67
45	1080.87	1080.88	1080.90	1080.87
50	1083.06	1083.03	1083.09	1083.06
55	1085.24	1085.19	1085.28	1085.24
60	1087.42	1087.35	1087.46	1087.43
65	1089.60	1089.50	1089.63	1089.60
70	1091.78	1091.66	1091.80	1091.78
75	1093.95	1093.82	1093.96	1093.95
80	1096.12	1095.97	1096.12	1096.12
85	1098.28	1098.13	1098.27	1098.28
90	1100.44	1100.28	1100.42	1100.44
95	1102.59	1102.44	1102.56	1102.59
100	1104.74	1104.60	1104.70	1104.74
105	1106.88	1106.75	1106.83	1106.88
110	1109.01	1108.90	1108.96	1109.01
120	1113.26	1113.22	1113.20	1113.25
140	1121.65	1121.85	1121.63	1121.65
160	1129.89	1130.47	1129.98	1129.89

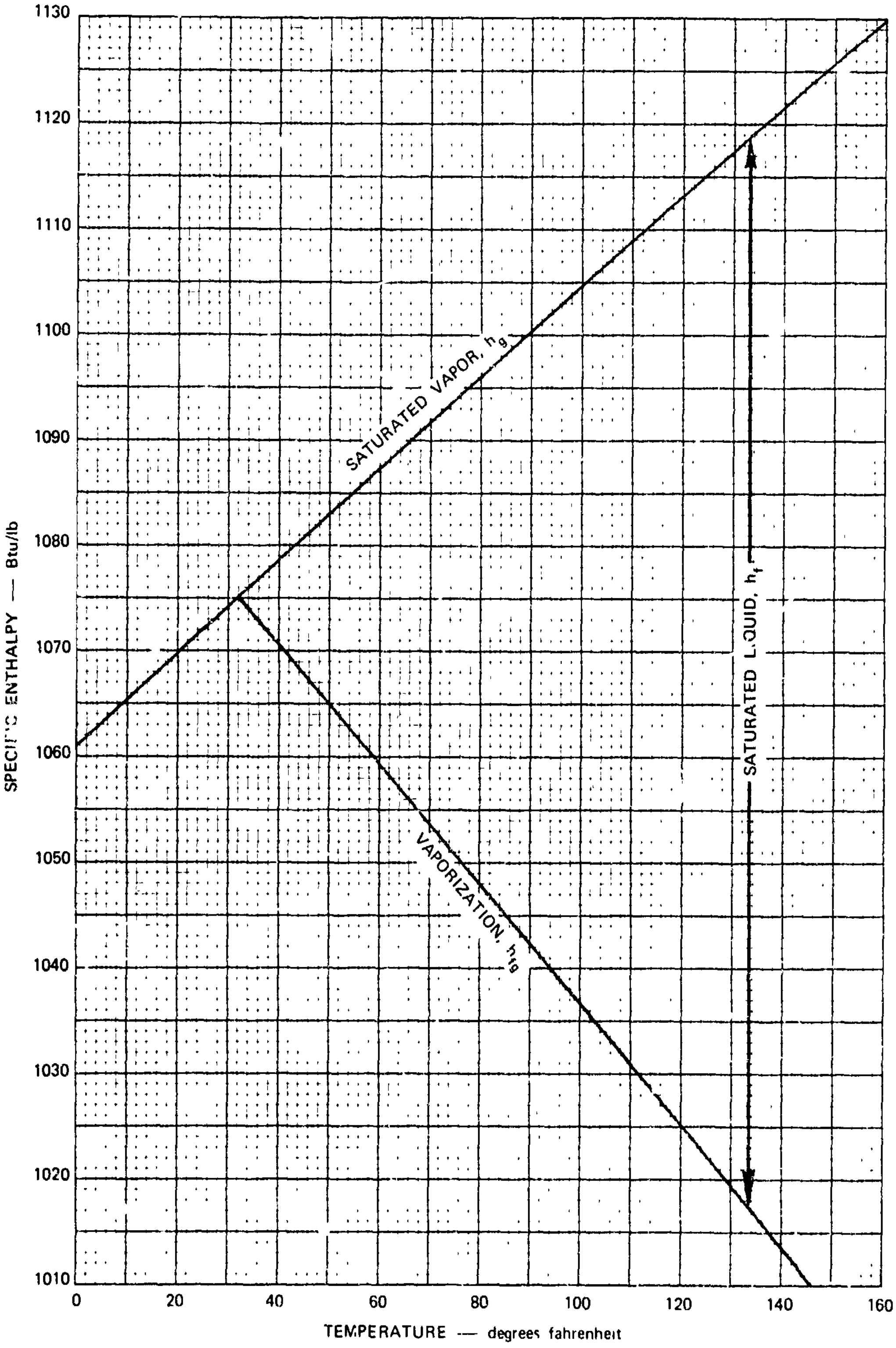

FIGURE 4 INTERPHASE ENTHALPIES OF WATER

Table 4

ABSOLUTE PRESSURE OF WATER AT SATURATION

TEMPERATURE t (F)	VAPOR PRESSURE p_{ws} (Lbf)/(SqIn)	TEMPERATURE t (F)	VAPOR PRESSURE p_{ws} (Lbf)/(SqIn)
-40	0.001861	70	0.36304
-20	0.006181	75	0.42979
-10	0.01082	77.5	0.46701
- 5	0.01419	80	0.50701
0	0.01849	82.5	0.54996
5	0.02396	85	0.59604
10	0.03087	87.5	0.64545
15	0.03957	90	0.69838
20	0.05045	92.5	0.75504
25	0.06400	95	0.81564
30	0.08080	97.5	0.88041
32 *	0.08858	100	0.94959
32	0.08859	105	1.1021
35	0.09991	110	1.2754
40	0.12164	115	1.4717
45	0.14746	120	1.6933
50	0.17799	130	2.2237
55	0.21397	140	2.8900
60	0.25618	160	4.7424
65	0.30554		

* Vapor pressure over ice.

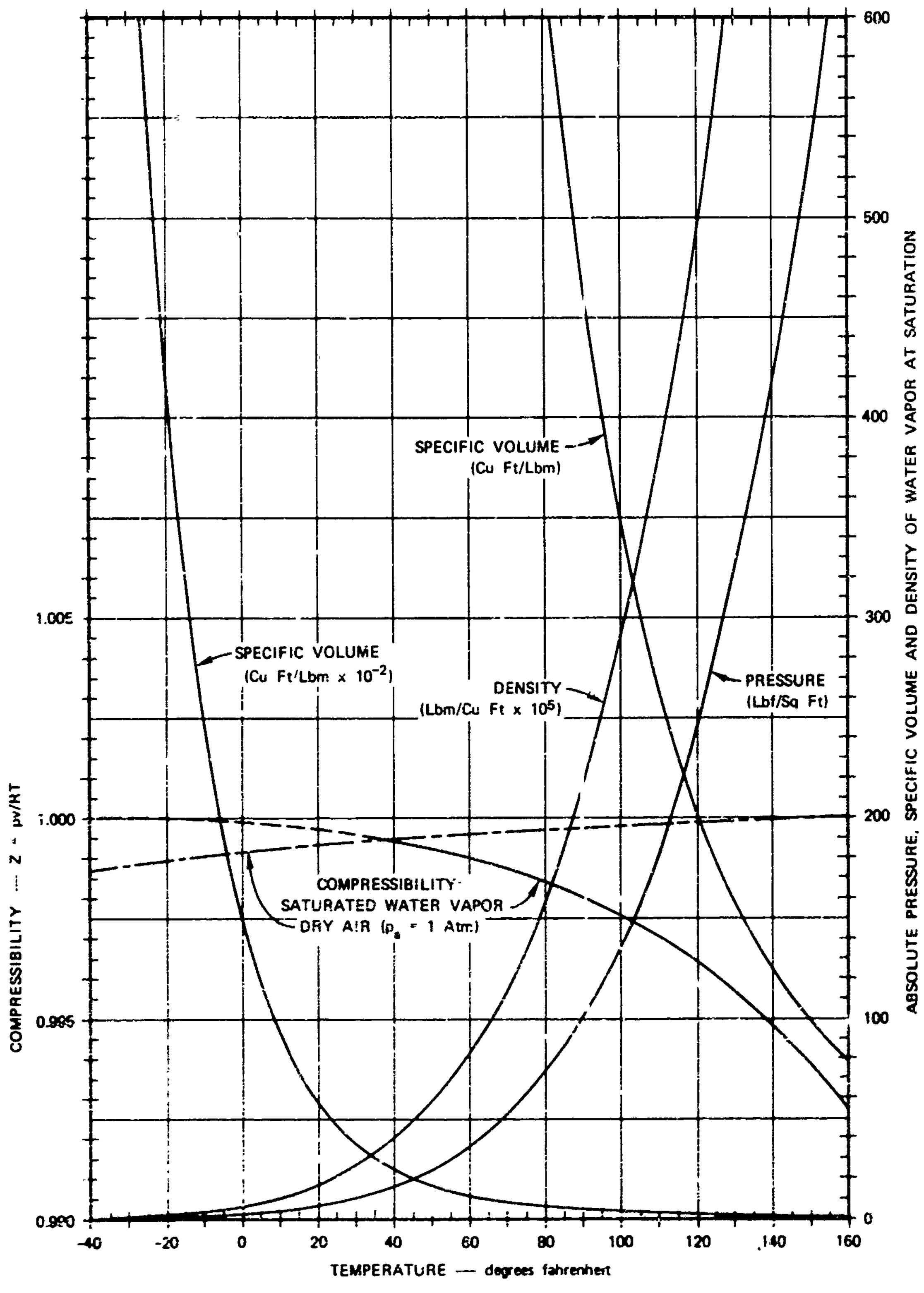

FIGURE 5 PHYSICAL PROPERTIES OF SATURATED WATER VAPOR
(Goff and Gratch Formulation)

$$
\begin{aligned}
p_{ws} = {} & 0.0191771 + (1.29380 \times 10^{-3})t \\
& + (1.20065 \times 10^{-5})t^2 + (4.54563 \times 10^{-7})t^3 \\
& + (6.86529 \times 10^{-11})t^4 + (2.19532 \times 10^{-11})t^5.
\end{aligned}
\tag{65}
$$

Range of application: +32°F to +160°F.

$$
\begin{aligned}
\mathrm{Ln}(p_{ws}) = {} & -3.81499 + (4.67990 \times 10^{-2})t \\
& -(1.10250 \times 10^{-4})t^2 + (2.08287 \times 10^{-7})t^3 \\
& -(2.24416 \times 10^{-10})t^4.
\end{aligned}
\tag{66}
$$

Range of application: +32°F to + 160°F.

$$
\begin{aligned}
p_{ws} = {} & 0.018493 + (9.69012 \times 10^{-4})t \\
& + (2.31795 \times 10^{-5})t^2 + (3.34982 \times 10^{-7})t^3 \\
& + (3.41669 \times 10^{-9})t^4 + (2.26351 \times 10^{-11})t^5.
\end{aligned}
\tag{67}
$$

Range of application: -40°F to + 32°F.

$$
\begin{aligned}
\sqrt{p_{ws}} = {} & 0.135979 + (3.56150 \times 10^{-3})t \\
& + (3.88253 \times 10^{-5})t^2 + (2.21018 \times 10^{-7})t^3 \\
& + (6.42695 \times 10^{-10})t^4.
\end{aligned}
\tag{68}
$$

Range of application: -40°F to +32°F.

Let $P = \mathrm{Ln}(p_{ws}) = 2.302585 \log_{10}(p_{ws})$. Then

$$t = 101.729 + 33.4599\, P + 2.24292\, P^2 + 0.127305\, P^3$$

Range of application: +32°F to +120°F.

$$0.08859 \le p_{ws} \le 1.6933 \quad \text{(Lbf/Sq in)}. \tag{69}$$

Let $S = \sqrt{p_{ws}}$. Then

$$t = -84.104 + 1430.8\ S - 12148.7\ S^2 + 72483.6\ S^3 - 260552.\ S^4 + 507574.\ S^5 - 410722.\ S^6$$

Range of application: -40^0F to $+32^0F$.

$$0.001861 \le p_{ws} \le 0.08858 \quad (Lbf)/(Sq\ in). \qquad (70)$$

IV REAL AND IDEAL PROPERTIES OF MOIST AIR

All known methods of psychrometric analysis are based on the equation of state for perfect gases, with or without corrections. For barometric pressures that approximate one standard atmosphere, optimum values for the properties of most air have been tabulated.[12] In these tables, corrections have been made for mixing interactions, for real gas properties of water vapor, and for real gas properties of dry air that contains little or no carbon dioxide.[9] For a generalized analytical procedure, it is practical and convenient to assume the validity of ideal gas relationships and Dalton's law of partial pressures. However, to establish a degree of confidence in the method, the magnitude of deviations should be examined.

Molecular Weights of Dry and Moist Air

Moist air is usually treated analytically as a binary mixture of dry air and water vapor, and the dry air fraction is assumed to have the composition shown in Table 2. With this composition, the apparent molecular weight of the dry-air fraction is 28.966, and the apparent molecular weight of moist air varies only with the relative amount of water vapor present in a mixture. In shelters, however, carbon dioxide concentrations may increase to about 0.50 percent by volume if ventilating rates are minimal and in a closed environment may be limited to about 2.00 percent by volume.

This increase in carbon dioxide concentration is caused by oxidation of food derivatives in metabolic processes that convert the energy essential to life. The volumetric ratio of carbon dioxide produced to oxygen consumed depends largely on diet and the presence or absence of

metabolic disorders. This ratio is the respiratory quotient, RQ. In general, respiratory quotients associated with dietary carbohydrates, proteins, and fats are about 1.00, 0.80, and 0.70, respectively.[14] A respiratory quotient of 0.82 is representative for a "normal" mixed diet.[15] With a respiratory quotient of 1.00, one unit volume of carbon dioxide is added to the air for each unit volume of oxygen extracted. The relative volumes of inert gases--nitrogen and argon--do not change, and the sum of oxygen and carbon dioxide concentrations is constant.[16] For approximating the effects of carbon dioxide on the molecular weight of dry and moist air, it is therefore convenient to assume that the respiratory quotient is unity.

When relative volumes of the inert gases--nitrogen and argon--correspond to those shown in Table 2, they behave essentially as a single gas with a molecular weight of 28.1564 units of mass per mass mole. Thus, when RQ = 1.00, the volumetric composition of dry air is

$$X_I = 0.7809 + 0.0093 = 0.7902$$

$$X_o + X_c = 0.2095 + 0.0003 = 0.2098$$

where

X_I = mole fraction of inert gases, nitrogen and argon

X_o = mole fraction of oxygen, and

X_c = mole fraction of carbon dioxide.

Then, the molecular weight of dry air as a function of carbon dioxide concentration is

$$\begin{aligned} M_a &= 0.7902\ (28.1564) + (0.2098 - X_c)\ (32.00) + 44.011\ X_c \\ &= 28.9628 + 12.011\ X_c . \end{aligned} \qquad (71)$$

For any value of M_a, as determined by Equation 71, the metabolic weight, M, of moist air is

$$M = X_w M_w + (1 - X_w) M_a = 18.016 X_w + (1 - X_w) M_a \qquad (72)$$

or

$$M = \frac{M_w M_a (1 + W)}{M_w + M_a W} = \frac{18.016 M_a (1 + W)}{18.016 + M_a W} \qquad (73)$$

where

X_w = mole fraction of water vapor and

W = humidity ratio in pounds of moisture per pound of dry air.

Within limitations of ideal gas relationships, $X_w = p_w/p$.

Apparent molecular weights of dry and moist air, as determined by Equations 71, 72, and 73, are shown on Figure 6 for the probable range of parameters. These parameters include (1) carbon dioxide concentration in dry air fraction, (2) ratio of vapor pressure to total pressure (or mole fraction of water vapor), and (3) humidity ratio. Values of dew-point temperature based on total pressure of one atmosphere (14.696 lbf/sq in) are also shown.

Real gas deviations from the ideal gas equation are often defined by a compressibility factor, $Z = p\bar{v}/\bar{R}T$, in which $\bar{v}$ is the molal volume. The effects of an erroneous molecular weight can be stated as a pseudo-compressibility factor, Z', defined by the following equation:

$$pv = \bar{R}T/M_c = Z'\bar{R}T/M_3$$

or

$$Z' = \frac{M_3}{M_c} \qquad (74)$$

in which

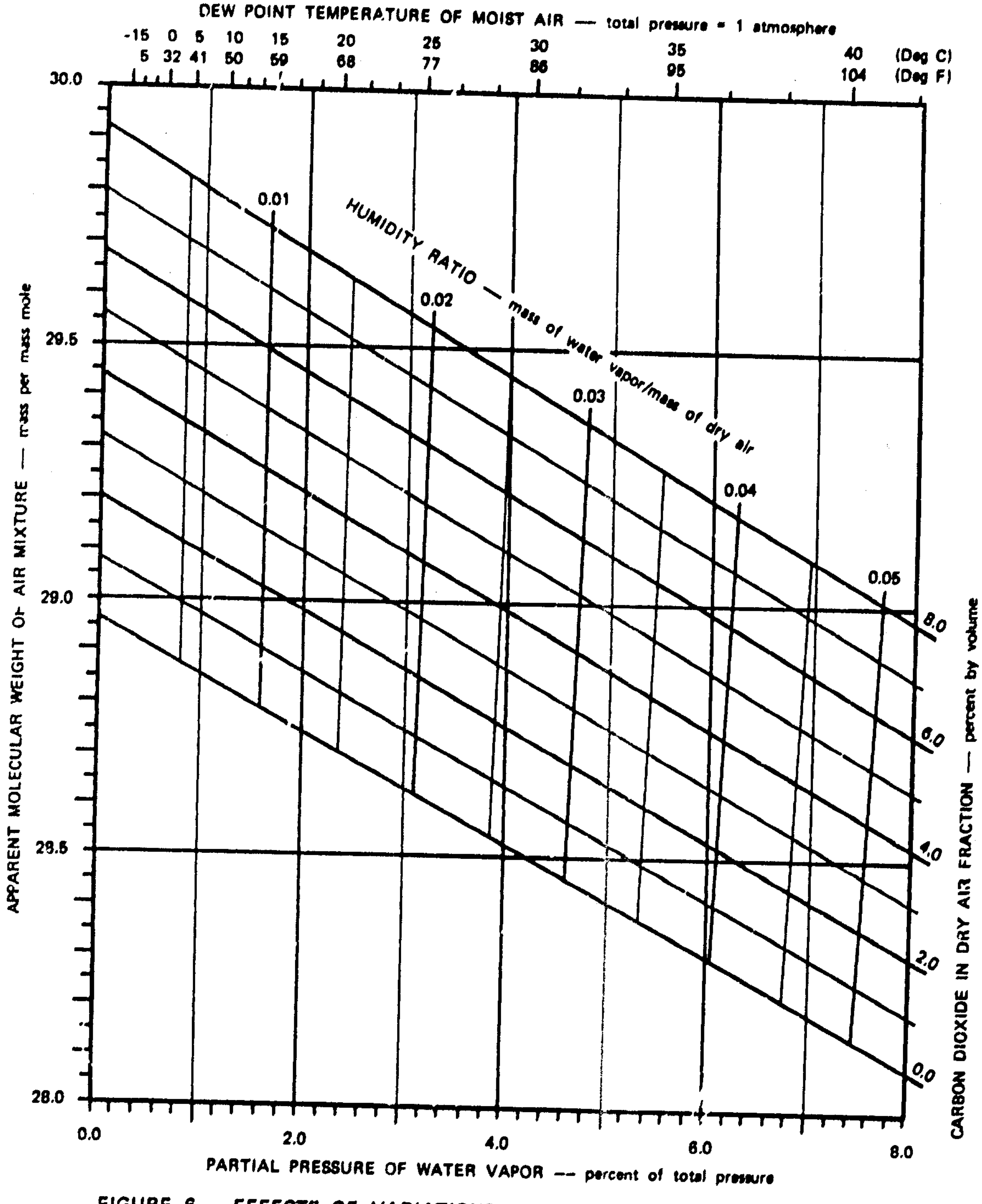

FIGURE 6 EFFECTS OF VARIATIONS IN WATER VAPOR AND CARBON DIOXIDE ON FUNDAMENTAL PARAMETERS FOR AIR MIXTURES

M_3 = 28.9664 = molecular weight of standard dry air having a carbon dioxide concentration of 0.03 percent by volume and

M_c = molecular weight of dry air having any probable carbon dioxide concentration.

Values of this pseudo-compressibility factor are given in Table 5 for subsequent comparison. These factors do not include the substantial deviations associated with real gas properties of carbon dioxide.

Table 5

PSEUDO-COMPRESSIBILITY FACTORS FOR DRY AIR

Carbon Dioxide Concentration, X_c (Percent by volume)	Pseudo-Compressibility Factor, Z'
0.00%	1.00012
0.03	1.00000
0.10	0.99971
0.20	0.99930
0.50	0.99805
1.00	0.99599
2.00	0.99190
4.00	0.98381

Deviations from Ideal Gas Relationships

The equations of state for ideal gases do not precisely define the properties of real gases or gas mixtures over a wide range of parameters. In general, the deviations tend to be small when the pressure is low or the temperature is high relative to the corresponding pressure or temperature at the critical point. Values of critical constants are given in Table 6 for major constituents of moist air.[13,11]

Table 6

CRITICAL CONSTANTS FOR COMPONENTS OF MOIST AIR

Constituent Gas or Vapor	Critical Temperature (°F)	Critical Pressure (Atm)	Critical Density (Lbm/cu ft)
Dry air	-221.3	37.2	21.7
Nitrogen	-232.8	33.5	19.4
Argon	-187.6	48.1	33.1
Oxygen	-181.8	49.7	26.8
Carbon dioxide	88.0	73.0	28.7
Water vapor	705.3	218.2	19.8

For conditions under which these gases are normally present in the atmosphere or in a habitable environment, partial pressures are low relative to critical pressures. State point temperatures are quite high relative to critical temperatures for all constituents except carbon dioxide and water vapor. However, partial pressures caused by carbon dioxide and water vapor are usually less than one-tenth of the total pressure.

Volumetric Deviations

Precise data on the thermodynamic properties of moist air at a total pressure of one atmosphere have been tabulated for a wide range of temperatures.[12] The basic, semiempirical, virial equation used in development of these data is the following pressure-volume-temperature relationship:[17]

$$p\bar{v} = \bar{R}T - \left[X_a^2 A_{aa} + 2\,X_a X_w\,A_{aw} + X_w^2 A_{ww}\right] p - \left[X_w^3 A_{www}\right] p^2 \qquad (75)$$

in which

p = barometric pressure (atmospheres),

$\bar{v}$ = molal volume (cu cm)/(gm mole),

$\bar{R}$ = universal gas constsnt,

= 82.054 (cu cm)(atm)/(gm mole)(^{0}K),

T = 273.15+t = absolute temperature (^{0}K),

X_a = mole fraction of dry air

X_w = mole fraction of water vapor,

A_{aa} = second virial coefficient for dry air,

A_{aw} = second virial coefficient for mixing interactions,

A_{ww} = second virial coefficient for water vapor, and

A_{www} = third virial coefficient for water vapor.

All of these virial coefficients are temperature dependent, and values have been tabulated.[9]

For a total pressure of one atmosphere, the pressure element vanishes, and Equation 75 becomes an explicit expression for molal volume, $\bar{v}$. The perfect gas counterpart for this special case is

$$\bar{v} = \bar{R}T \qquad (76)$$

In Equation 75, it is apparent that the term containing A_{aa} determines deviations due to dry air, the term containing A_{aw} determines deviations

due to mixing interactions, and the terms containing A_{ww} and A_{www} determine deviations due to water vapor.

Total and partitioned volumetric deviations in parts per million, as determined by Equations 75 and 76, are shown in Table 7 for ranges of temperature and humidity that might occur in a shelter environment. Negative values indicate that the actual volume is less than a volume determined by the ideal gas equation. Ideal molal volumes determined by Equation 76 vary from 22,143 at $0^{o}C$ to 28,977 at $80^{o}C$, in units of cu cm/gm mole.

Within the range of parameters covered by Table 7, partial pressures due to dry air are at least 90 percent of the total pressure, and the mass of water vapor is less than 7 percent of the total mass of moist air. This largely accounts for the relatively large contribution made by dry air to total deviations at temperatures of $100^{o}F$ or less. At higher temperatures, deviations due to dry air are greatly reduced as its compressibility approaches unity. Deviations due to water vapor and mixing interactions decrease slowly as temperature increases, but increase rapidly as the mole fraction of water vapor increases. All of the tabulated deviations in volume are relatively small.

Curves were plotted on Figure 5 for the specific volume of saturated water vapor, using accurate tabulated values.[12] Comparable values can be determined with the ideal gas equation, $V_w = R_w T/p$, at corresponding temperatures and pressures. For example, tabulated values for a specific volume of saturated water vapor are less than calculated ideal values by the amounts, 0.85 and 0.64 cu ft/lbm, at temperatures of $100^{o}F$ and $140^{o}F$, respectively. These deviations are barely detectable at the scales with which the curves are plotted in Figure 5. Within any reasonable range of conditions that might develop in shelters, the basic equation of state for perfect gases provides a good definition of pressure-volume-temperature relationships, for water vapor as well as air.

Table 7

MOIST-AIR DEVIATIONS FROM IDEAL GAS RELATIONSHIP

(Barometric Pressure = 1 Atmosphere)

TEMPERATURE		WATER VAPOR IN AIR		DEW POINT	VOLUMETRIC DEVIATIONS FROM IDEAL GAS LAW (Parts Per Million)				COMPRESSIBILITY OF MOIST AIR
^{o}C	^{o}F	MOLE FRACTION x_w	HUMIDITY RATIO W	^{o}F	DUE TO DRY AIR	DUE TO MIXING REACTIONS	DUE TO WATER VAPOR	TOTAL DEVIATION	$Z = p\bar{v}/\bar{R}T$
0	32	0.00	0.00000	--	-589	0	0	-589	0.99941
10	50	0.00	0.00000	--	-469	0	0	-469	0.99953
		0.01	0.00628	44.8	-460	-34	-6	-500	0.99950
20	68	0.00	0.00000	--	-366	0	0	-366	0.99963
		0.01	0.00628	44.8	-359	-30	-5	-394	0.99961
		0.02	0.01269	63.8	-352	-60	-21	-433	0.99957
30	86	0.00	0.00000	--	-276	0	0	-276	0.99972
		0.01	0.00628	44.8	-271	-27	-4	-302	0.99970
		0.02	0.01269	63.8	-265	-54	-17	-337	0.99966
		0.04	0.02592	84.4	-254	-107	-69	-430	0.99957
40	104	0.00	0.00000	--	-198	0	0	-198	0.99980
		0.01	0.00628	44.8	-194	-25	-4	-223	0.99978
		0.02	0.01269	63.8	-190	-49	-14	-254	0.99975
		0.04	0.02592	84.4	-183	-97	-57	-337	0.99966
		0.06	0.03970	97.4	-175	-142	-129	-446	0.99955
50	122	0.00	0.00000	--	-130	0	0	-130	0.99987
		0.01	0.00628	44.8	-127	-23	-3	-153	0.99985
		0.02	0.01269	63.8	-125	-45	-12	-182	0.99982
		0.04	0.02592	84.4	-120	-88	-48	-256	0.99974
		0.06	0.03970	97.4	-115	-129	-109	-353	0.99965
		0.08	0.05408	107.0	-110	-169	-194	-473	0.99953
		0.10	0.06911	114.8	-105	-206	-303	-614	0.99939
60	140	0.00	0.00000	--	-70	0	0	-70	0.99993
		0.01	0.00628	44.8	-68	-21	-3	-92	0.99991
		0.02	0.01269	63.8	-67	-41	-10	-118	0.99988
		0.04	0.02592	84.4	-64	-80	-42	-186	0.99981
		0.06	0.03970	97.4	-62	-118	-94	-273	0.99973
		0.08	0.05408	107.0	-59	-154	-167	-380	0.99962
		0.10	0.06911	114.8	-57	-188	-262	-506	0.99949
70	158	0.00	0.00000	--	-17	0	0	-17	0.99998
		0.01	0.00628	44.8	-17	-19	-2	-38	0.99996
		0.02	0.01269	63.8	-17	-37	-9	-63	0.99994
		0.04	0.02592	84.4	-16	-73	-36	-125	0.99988
		0.06	0.03970	97.4	-15	-107	-80	-203	0.99980
		0.08	0.05408	107.0	-15	-140	-143	-298	0.99970
		0.10	0.06911	114.8	-14	-171	-225	-410	0.99959
80	176	0.00	0.00000	--	29	0	0	29	1.00003
		0.01	0.00628	44.8	28	-17	-2	9	1.00001
		0.02	0.01269	63.8	28	-34	-8	-14	0.99999
		0.04	0.02592	84.4	27	-67	-31	[illegible]	0.99993
		0.06	0.03970	97.4	26	-98	-70	-142	0.99986
		0.08	0.05408	107.0	24	-128	-124	-228	0.99977
		0.10	0.06911	114.8	23	-156	-194	-327	0.99967

The values of compressibility in Table 7 can be compared with the pseudo-compressibility factors in Table 5, which relate to the molecular weights of dry air with various carbon dioxide concentrations. It is obvious that the effects of carbon dioxide concentration on molecular weight of dry air, M_a, are potentially more significant than deviations associated with the ideal gas equation. When per capita rates of ventilation are low, appropriate mean values of M_a could well be used in lieu of the standard value, 28.966, for atmospheric air.

Enthalpy Deviations

Tabulated properties of moist air include accurate values for specific enthalpies of dry and moist air, when saturated at a barometric pressure of one atmosphere.[12] Moist air enthalpies were determined by a virial equation that is thermodynamically related and similar in form to Equation 75.[17] The treatment of enthalpy deviations will be limited to comparisons among real and ideal properties of moist air and its components, dry air and water vapor.

In Equations 52 and 53, specific enthalpy of moist air is defined as the sum of enthalpies for one pound of dry air and the contained water vapor. This is consistent with thermodynamic relationships for ideal gases. It is likewise consistent to assume that the values of specific heat and enthalpy are functions only of temperature.

Average values for the specific heat of dry air at constant pressure can be determined for any selected temperature range from tabulated values for enthalpy of dry air.[12] For temperature ranges of 0° to 40°F, 0 to 80°F and 0 to 120°F, average values of c_{pa} are, respectively, 0.24020, 0.24026, and 0.24034 in (Btu/(lbm)(°F). From another source, interpolated values for specific heats of dry air and water vapor at constant low pressures are shown in Table 8.[18]

Table 8

SPECIFIC HEATS OF DRY AIR AND WATER VAPOR

Temperature (°F)	Specific Heat at Constant Pressure (Btu)/(lbm)(°F) Dry Air	Water Vapor
-40	0.2393	0.4423
0	0.2394	0.4430
40	0.2396	0.4439
80	0.2398	0.4451
120	0.2401	0.4465
160	0.2405	0.4482

Within the ranges of temperature and vapor pressure that are likely to occur in a habitable environment, it is reasonable and convenient to use constant values for specific heat; c_{pa} = 0.240 for dry air and c_{pw} = 0.446 for water vapor. In general, the specific heat of water vapor at constant pressure will be well within limits of 0.44 to 0.46 (Btu/(lbm)(°F), and values higher than 0.45 would be associated with high effective temperatures or storage of metabolic heat.[19]

In Equation 52, the specific enthalpy of water vapor in moist air is evaluated by the term, $h_w = Wh_g$ (Btu)/(lbm of dry air). The humidity ratio, W, is the mass of water vapor in one pound of dry air, and the quantity, h_g, is the specific enthalpy of saturated water vapor at the mixture temperature. Since water vapor in moist air is usually superheated rather than saturated, this term in the equation implies that the specific enthalpy of superheated water vapor is identical to the specific enthalpy of saturated water vapor at the same temperature. This implication is

consistent for ideal gases, but is a reasonable approximation only for water vapor at low pressures or degrees of superheat. The nature of this deviation is shown in Figure 7, which is a pressure enthalpy chart derived from a recent table for properties of superheated steam.[20] Unfortunately, the table does not include data for temperatures less than 32°F. On Figure 7, the lines of constant temperature are not quite parallel to the grid lines for constant specific enthalpy. At any temperature, the specific enthalpy of superheated water vapor is somewhat higher than the specific enthalpy of saturated water vapor at the same temperature.

Since the constant temperature lines on Figure 7 are straight, the apparent deviation in specific enthalpy of water vapor varies linearly with pressure and would be maximum at zero pressure. However, the resultant deviation in specific enthalpy of moist air is the product, $W(h_u - h_g)$, where h_u is the specific enthalpy of superheated water vapor. By combining Equations 45 and 50, another equation is obtained for humidity ratio.

$$W = \frac{\phi}{\frac{p}{p_{ws}} - \phi} \left(\frac{M_w}{M_a} \right) \tag{77}$$

It can then be shown that the maximum deviation in specific enthalpy of moist air would occur when the relative humidity, ϕ', or degree of saturation, μ', is

$$\phi' = 1 - \mu' = \frac{p}{p_{ws}} - \sqrt{\frac{p}{p_{ws}} \left(\frac{p}{p_{ws}} - 1 \right)} \tag{78}$$

The relative magnitude of this deviation in enthalpy of superheated water vapor can be illustrated as follows:

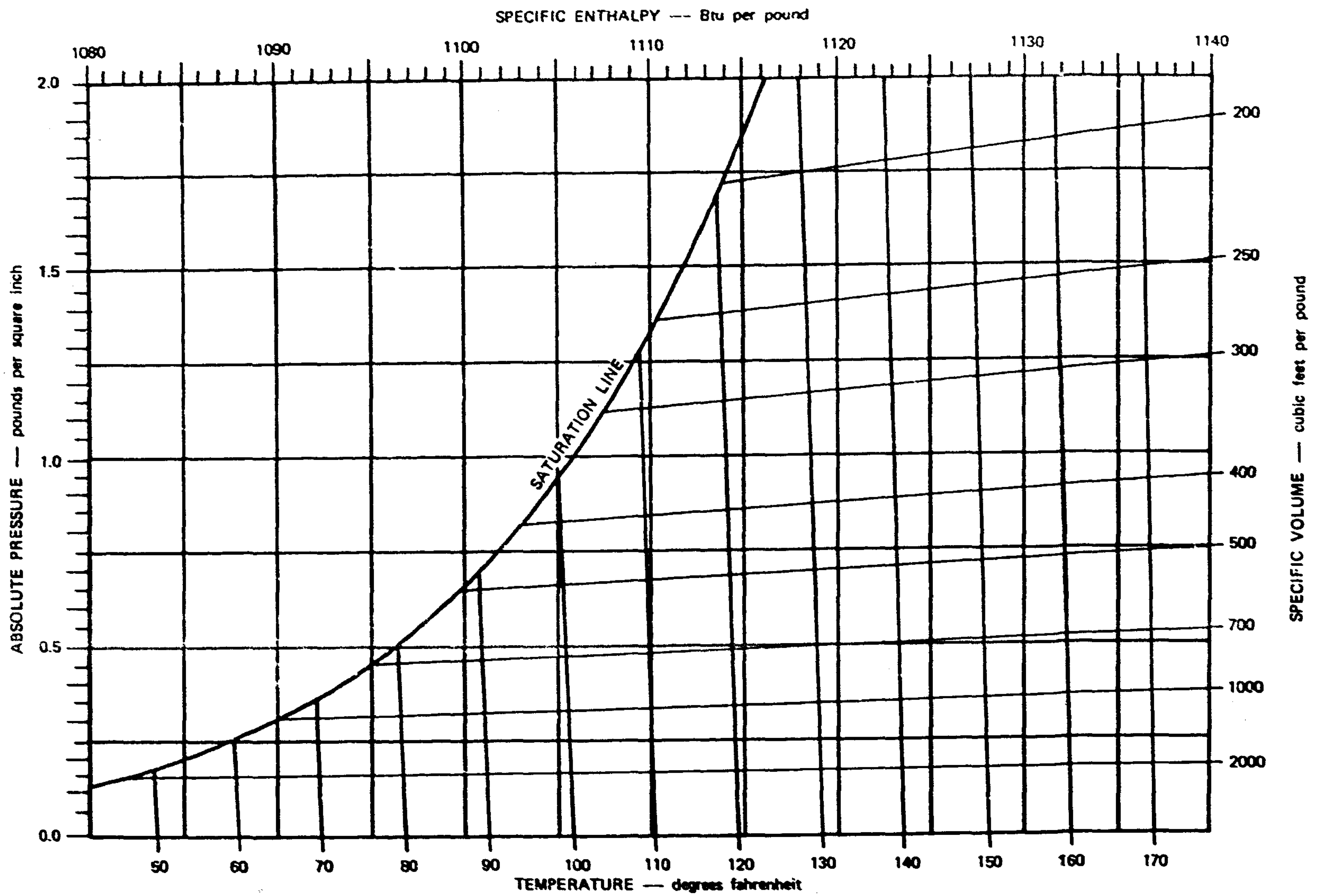

FIGURE 7 PRESSURE - ENTHALPY DIAGRAM FOR WATER VAPOR
(Data from ASME Steam Tables - 1967)

Example: A sample of moist air has a total pressure of one atmosphere and a temperature of 100°F. What is the maximum effect of superheat deviations on the specific enthalpy of moist air? What is the corresponding effect of superheat deviation upon average specific heat of water vapor?

Values of pertinent parameters are:[20]

Saturated vapor pressure, p_{ws} = 0.94924

Pressure ratio, p/p_{ws} = 15.482

From Equation 78, for maximum effect, $\phi' = 0.50835$ and $\mu' = 0.49165$

Superheated vapor pressure, $p_w = (0.50835)\ (0.94924) = 0.48255$

Saturated vapor enthalpy, $h_g = 1105.1$

Superheated vapor enthalpy, $h_u = 1105.4$

Enthalpy deviation, $h_u - h_g = 0.3$ (Btu/lbm)

Dew point, $t_d = 78.5\ ^oF$

Dew point depression, $t-t_d = 21.5^oF$

Enthalpy at dew point, $h_{gd} = 1095.8$

By Equation 45, the humidity ratio is

$$W = \frac{(0.48255)\ (0.62197)}{14.696 - 0.48255} = 0.02112\ .$$

Deviation in specific enthalpy of moist air is

$$W\ (h_u - h_g) = 0.00633\ \text{(Btu)/(lbm of dry air)}.$$

From Equation 52, the uncorrected specific enthalpy of moist air is

$h = (0.24 \times 100) + (0.02112 \times 1105.1) = 47.340$ (Btu)/(lbm of dry air).

Then, the corrected value for specific enthalpy of moist air is

$h' = 47.340 + 0.00633 = 47.346$ (Btu)/(lbm of dry air).

The difference is small at 100°F, and is probably negligible for all conditions associated with habitable environments.

If Equations 52 and 53 are solved simultaneously for the apparent average value of c_{pw}, the specific heat of water vapor at constant pressure is

$$c_{pw(apparent)} = \frac{h_g - h_{gd}}{t - t_d} . \tag{79}$$

Then, if a correction term is added to Equation 52,

$$h = c_{pa}t + Wh_g + W(h_u - h_g) = c_{pa}t + Wh_u \tag{80}$$

and Equations 80 and 53 are solved for the actual average value of c_{pw},

$$c_{pw(actual)} = \frac{h_u - h_{gd}}{t - t_d} . \tag{81}$$

By introducing numerical values for the example, the apparent average value of specific is

$$c_{pw(apparent)} = \frac{1105.1 - 1095.8}{21.5} = 0.4326 \text{ (Btu)/(lbm)(}^\circ\text{F)}$$

and the actual value is

$$c_{pw(actual)} = \frac{1105.4 - 1095.8}{21.5} = 0.4465 \text{ (Btu)/(lbm)(}^\circ\text{F)} ,$$

Thus, the low values for specific heat of water vapor associated with polynomial approximations for enthalpy of saturated water vapor, as defined by Equation 57, are due to the enthalpy deviations for superheated water vapor shown in Figure 7. Equations 52 and 53 are compatible only when the apparent value of c_{pw} determined by coefficients of the polynomial approximation is used.

Wet-Bulb Temperature and Pressure Relationships

In contrast to the psychrometer wet-bulb temperature, the thermodynamic wet-bulb temperature has a well-defined relationship with the more fundamental properties of moist air: dry-bulb temperature, specific enthalpy, humidity ratio, and vapor pressure. Wet-bulb temperature readings taken with a sling or aspirating psychrometer are sensitive to several factors, including radiation, air velocity, water purity, and dimensions of the sensor element. To obtain measurements that closely approach the thermodynamic wet-bulb temperature, high quality apparatus and appropriate techniques are necessary.[21]

Thermodynamic wet-bulb temperature, t^*, can be defined as the temperature of saturated moist air that satisfies the following energy equation:[1]

$$h - Wh^*_\omega = h_a + W(h_g - h^*_\omega) = h^*_s - W^*_s h^*_\omega = M^* \tag{82}$$

in which

h = specific enthalpy of moist air,

h_a = specific enthalpy of dry air,

h_g = specific enthalpy of saturated water vapor,

$h^*_f \approx h^*_\omega$ = specific enthalpy of condensed water at t^*

h^*_s = specific enthalpy of saturated air at t^*,

W = humidity ratio of moist air,

W^*_s = humidity ratio of saturated air at t^*, and

M^* = a function of moist air properties at wet-bulb temperature, sometimes called "Sigma Heat."[22] Values have been tabulated for a barometric pressure of one atmosphere.[1]

Equation 82 provides a relationship for evaluating basic properties of moist air when reliable values of wet-bulb temperature and either dry-bulb

temperature or humidity ratio are known. In general, thermodynamic wet-bulb temperature is a convenient parameter only for graphical methods of analysis.

There are two useful empirical equations for determining vapor pressures directly from known dry-bulb and wet-bulb temperature data. The Carrier equation is[23]

$$p_w = p^*_{ws} - \frac{(p - p^*_{ws})(t - t^*)}{2831 - 1.43\, t^*} \qquad (83)$$

and the Ferrel equation is[24]

$$p_w = p^*_{ws} - 0.000367\, p(t - t^*) \left(1 + \frac{t^* - 32}{1571}\right) \qquad (84)$$

In these equations

p_w = vapor pressure of moist air,

p^*_{ws} = saturated vapor pressure at wet-bulb temperature,

p = barometric pressure,

t = dry-bulb temperature (°F), and

t^* = wet-bulb temperature (°F).

Values of vapor pressure, p_w, obtained by means of Equations 82, 83, and 84 are compared in Table 9. Corresponding dew points were determined by interpolation in steam tables.[12] Vapor pressures based on thermodynamic wet-bulb relations can be used as a standard for comparison. Values obtained with the Carrier equation conform more closely to this "standard" than values obtained with the Ferrel equation.

The last four columns in Table 9 compare vapor pressures and dew points obtained by means of Equations 46 and 51, which relate vapor pressure to humidity ratio. These equations can be written in the forms

Table 9

DETERMINATION OF DEW POINTS AND PARTIAL PRESSURES OF WATER VAPOR AT STATE POINTS ON LINES OF CONSTANT EFFECTIVE TEMPERATURE

(Barometric Pressure, p = 14.696 Lbf/SqIn)

TEMPERATURES AT STATE POINT (t_e = Effective (°F), t = Dry Bulb (°F), t^* = Wet Bulb (°F))			RELATIVE HUMIDITY %	HUMIDITY RATIO Lbv/Lbda	COMPARATIVE VALUES OF VAPOR PRESSURE AND DEW POINT TEMPERATURE (p_w = Vapor Pressure, Lbf/SqIn) (t_d = Dew Point, Deg F)									
					BASED ON THERMODYNAMIC WET BULB RELATIONSHIPS		BASED ON "CARRIER" EQUATION		BASED ON "FERREL" EQUATION		BASED ON IDEAL GAS LAW AND HUMIDITY RATIO		BASED ON HUMIDITY RATIOS AND SATURATED VAPOR PRESSURE	
t_e	t	t^*	ϕ	W	p_w	t_d	p_w	t_d	p_w	t_d	p_w	t_d	p_w	t_d
50	51.0	39.0	29.3	0.00230	0.0540	21.40	0.0540	21.40	0.0520	20.62	0.0542	21.49	0.0541	21.47
68	68.0	68.0	100.0	0.01475	0.3390	68.00	0.3390	68.00	0.3390	68.00	0.3404	68.12	0.3390	68.00
	70.0	64.5	74.8	0.01175	0.2714	61.62	0.2714	61.62	0.2700	61.48	0.2725	61.73	0.2714	61.62
	72.0	61.0	53.4	0.00895	0.2076	54.17	0.2076	54.17	0.2050	53.83	0.2085	54.28	0.2076	54.16
	76.0	54.0	20.4	0.00388	0.0907	32.59	0.0905	32.54	0.0860	31.35	0.0911	32.69	0.0907	32.59
77	77.0	77.0	100.0	0.02016	0.4594	77.00	0.4594	77.00	0.4594	77.00	0.4614	77.13	0.4594	77.00
	82.0	72.0	62.1	0.01462	0.3361	67.75	0.3361	67.75	0.3332	67.50	0.3375	67.87	0.3360	67.74
	86.0	68.0	39.7	0.01057	0.2445	58.69	0.2445	58.69	0.2397	58.14	0.2456	58.81	0.2445	58.69
	95.0	59.0	7.1	0.00248	0.0581	22.96	0.0578	22.85	0.0497	19.68	0.0584	23.05	0.0581	22.95
	98.1	55.9	0.0	0.00000	0.0000	-	-0.0004	-	-0.0095	-	0.0000	-	0.0000	-
86	86.0	86.0	100.0	0.02731	0.6154	86.00	0.6154	86.00	0.6154	86.00	0.6181	86.14	0.6154	86.00
	91.5	83.0	70.2	0.02266	0.5142	80.43	0.5146	80.45	0.5116	80.27	0.5166	80.57	0.5143	80.43
	97.0	80.0	48.2	0.01830	0.4181	74.17	0.4182	74.18	0.4175	73.77	0.4200	74.31	0.4180	74.17
	106.0	74.0	19.7	0.01026	0.2376	57.89	0.2375	57.89	0.2274	56.67	0.2385	58.00	0.2373	57.86
95	128.0	87.0	20.0	0.01846	0.4216	74.42	0.4223	74.47	0.4064	73.33	0.4235	74.56	0.4212	74.39
100	144.0	94.0	16.6	0.02347	0.5320	81.47	0.5331	81.54	0.5106	80.21	0.5345	81.62	0.5313	81.43

$$\frac{p_w}{p} = \frac{W}{W + 0.62197} \tag{46}$$

and

$$\frac{p_w}{p_{ws}} = \frac{W(W_s + 0.62197)}{W_s(W + 0.62197)} \tag{51}$$

Although both of these equations are derived from the equation of state for ideal gases, Equation 51 includes tabulated properties of moist air at saturation and is more exact. However, Equation 46 is more convenient and is probably adequate for most purposes.

In the foregoing comparisons, deviations in the properties of moist air associated with ideal gas relationships are shown to be relatively small at temperatures less than about 160°F and humidity ratios less than 0.060 pounds of moisture per pound of dry air. Environmental conditions are likely to approach these limits only when the effects of fire prevail. One can therefore conclude that the ideal gas relationships provide a necessary, adequate and convenient basis for environmental analyses in connection with human habitation. The paramount concern then reverts to the rational evaluation of parameters for internal loads, ambient conditions, and physiological criteria.

V ENVIRONMENTAL CRITERIA AND METABOLIC PARAMETERS

Rational criteria for a thermal environment must be closely associated with capabilities for heat transfer and for regulation of heat transfer by evaporation, radiation, convection, and conduction between the human body and its surroundings. For prolonged exposure to extreme environmental states under these criteria, there must be a net capability for establishing a state of metabolic equilibrium at a safe body temperature. This is a necessary constraint, but the "safe body temperature" or time-tolerance effects are subject to wide variations among types of individuals.[25] Moreover, environmental states at the threshold of the noncompensable zone are associated with a high degree of heat stress. Under such conditions, any increase in physical activity or deficit in rate of water replacement may precipitate metabolic failure.[26] Superficial heat stress may be incipient when the skin becomes visibly moist or the pulse rate starts to increase.

In a hot environment, evaporation from skin surfaces is the dominant process for heat transfer, and this effect is limited by vapor pressure gradients at skin surfaces or by the rate of sweat secretion. Skin temperature, sweat rate, heart rate, and blood flow through superficial tissues progressively increase to maximize heat transfer and preserve critical body temperatures. Heat transfer rates are directly proportional to air velocity and inversely proportional to vapor pressure in air. In a warm environment, it is advantageous to wear little or no clothing and to avoid sleeping on an insulating pad or mattress.

In a cold environment, the effects of radiation and convection from clothing and conduction through clothing are dominant. Evaporative effects may be largely caused by moisture added to inspired air in the respiratory

tract. If necessary to preserve critical temperatures, the body responds to a cool environment in two ways: (1) by restricting blood flow through superficial tissues, thus increasing their insulating value, and (2) by inducing muscular tension, shivering, or greater physical activity, all of which increase metabolic rate and tend to warm the body. If these responses are inadequate, body core temperatures must fall. Chilling effects are accentuated by relatively low radiant temperatures or high air velocities. Within practical limits, a person can largely avoid these effects and maintain a substantially uniform metabolic rate or state of thermal equilibrium by wearing optimum clothing, i.e., clothing that is appropriate at any time to requirements for insulation in the prevailing environment.[3]

Many indexes have been devised to evaluate an environment with respect to physiological reactions or subjective responses resulting from exposure of test subjects under similar conditions. Some of these indexes consider only a few factors that affect the degree of comfort or heat stress; others are more comprehensive. In general, these factors relate to heat transfer and metabolic parameters such as air temperature, mean radiant temperature, skin temperature, vapor pressure, air motion, metabolic rate, clothing, and posture. To promote the development and establishment of improved indexes, recent emphasis has been placed on the correlation of empirical data with governing principles of heat transfer.[3,25,27] With reference to any particular index, environmental criteria should logically be based on a permissible degree of thermal stress between limits associated with comfort or survival and should be supported by realistic interpretation of probable time dependent physiological effects on a heterogeneous population.[25]

Effective Temperature

The effective temperature index evaluates the combined effects of temperature, humidity, and air motion on sensations of thermal comfort or discomfort. It has been used extensively as the basis for environmental criteria since 1930, with only minor changes.[28] Its limitations and deficiencies have long been recognized and often discussed.

Limitations on applicability of this index reflect the conditions and procedures used to obtain the basic data. Normally clothed, healthy young subjects moved alternately between two rooms, one warm and dry and the other relatively cool and humid. In each room, dry-bulb temperatures and mean radiant temperatures were virtually equal. Conditions in one room were changed slowly until the subjects could detect no difference between the two rooms with regard to thermal sensations; that is, the "effective temperatures" were equivalent. By repeating this experiment with the same subjects and with several combinations of temperature and humidity, data were obtained to construct on a psychrometric chart a line that intersected the saturation curve. The effective temperature along this line of constant sensory impressions was then assigned the value corresponding to the dry-bulb, wet-bulb, and dew-point temperatures at the point of intersection with the saturation curve.[29]

The major deficiency ascribed to the effective temperature index concerns the relative influence of humidity, which is overestimated in or below the comfort zone and underestimated at temperatures near the limit of human tolerance to heat stress.[29] This contention is supported by later information, which indicates generally that lines of equivalent environmental states differ significantly with respect to slope and linearity from lines of constant effective temperature.[25,26,27,28]

For environmental engineering analysis, an index that defines equivalent states should be compatible with conditions that would probably prevail in shelters during an emergency and should be reducible to a manageable equation in terms of convenient physical parameters. The environment in a shelter that has a basic ventilating system without supplementary cooling or heating apparatus is subject to wide seasonal and spatial variations. Air temperatures and mean radiant temperatures would often be quite different. Insofar as possible, the occupants--who would usually be a random sample of the general population--would wear clothing that is appropriate in the immediate environment. Exposure to conditions in the shelter would be prolonged. The effective temperature index was not derived from data obtained under such conditions, nor was any other known index. However, limiting criteria for warm environments in shelters are within the range of 80 to 85 FET, the least controversial range for application of the index to situations in which occupants are at rest and normally clothed, and air-wall temperatures are nearly equal.

A complete evaluation of the effective temperature index is presented in the form of a nomogram from which numerical values of effective temperature can be determined when values of dry-bulb and wet-bulb temperature are known.[28,25] For use in psychrometric analyses, however, vapor pressure is a more convenient parameter than wet-bulb temperature.

When determined geometrically from the ET nomogram, any specific value of effective temperature that may be selected for use as an environmental criterion is a linear function of dry-bulb and wet-bulb temperature. An equivalent expression that includes vapor pressure rather than wet-bulb temperature is needed.

Figure 8 is an abridged psychrometric chart constructed with rectangular coordinates of dry-bulb and wet-bulb temperatures. Lines of constant effective temperature are superimposed on this grid. Intersections of these ET lines with lines of constant relative humidity or lines

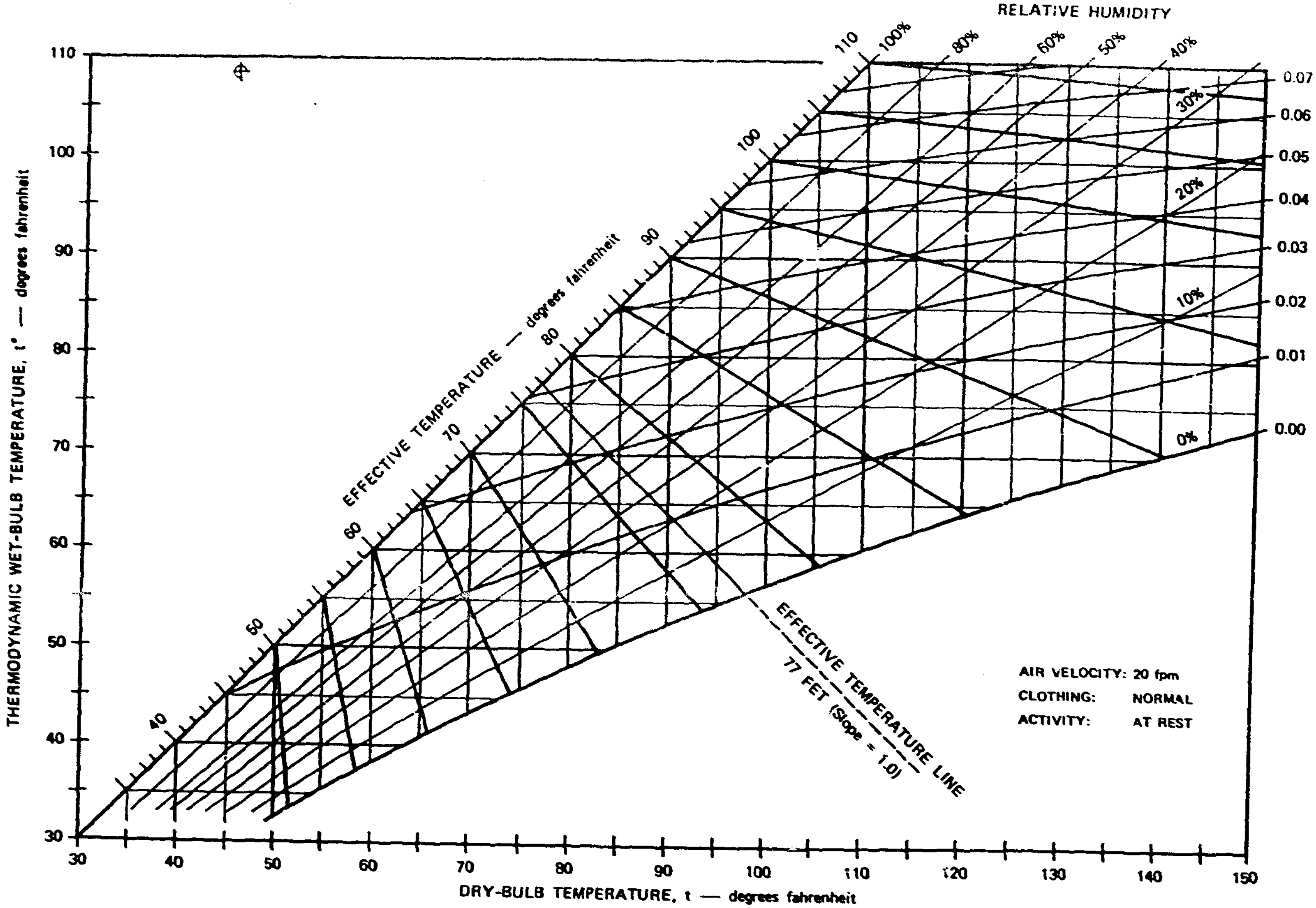

FIGURE 8 PSYCHROMETRIC PROPERTIES ASSOCIATED WITH EFFECTIVE TEMPERATURE INDEX

of constant vapor pressure determine values of dry-bulb temperature and vapor pressure along lines of constant effective temperature. If these values are plotted with rectangular coordinates of dry-bulb temperature and vapor pressure, the resultant effective temperature lines are quite straight, as shown in Figure 9. Departures from linearity are most apparent when relative humidities are less than 30 percent and effective temperatures are higher than 75°F, but such a combination of conditions will seldom occur in shelters. Therefore, slopes were determined for the portions of 27 effective temperature lines that lie above 30 percent relative humidity. The slope of each ET line was determined by two points. One point corresponds to the tabulated value of saturated vapor pressure, p_{wse}, at the effective temperature, t_e. The other point was determined by the mean values of vapor pressure and dry-bulb temperature at intersections of each ET line with the 30 and 40 percent relative humidity lines on a larger version of the chart shown in Figure 8. The mean of values at two points of intersection was used to reduce random errors. Vapor pressures were determined by the relationship, $p_w = \phi p_{ws}$, where ϕ is relative humidity and p_{ws} is the tabulated value of saturated vapor pressure at dry-bulb temperature, t. Pairs of points were determined for effective temperatures of 45, 50, 55, 60, 65, 68, 70, 72, 74, 75, 76, 77, 78, 79, 80, 81, 82, 83, 84, 85, 86, 87, 90, 95, 100, 105, and 110°F. Each pair of points determines a linear equation that relates vapor pressure to dry-bulb and effective temperatures. The general form of these equations is

$$p_w = p_{wse} - S(t - t_e) \tag{85}$$

where

t_e = effective temperature (°F),

t = dry-bulb temperature (°F),

p_w = vapor pressure in air (lb/sq in),

p_{wse} = saturated vapor pressure at t_e (lb/sq in)

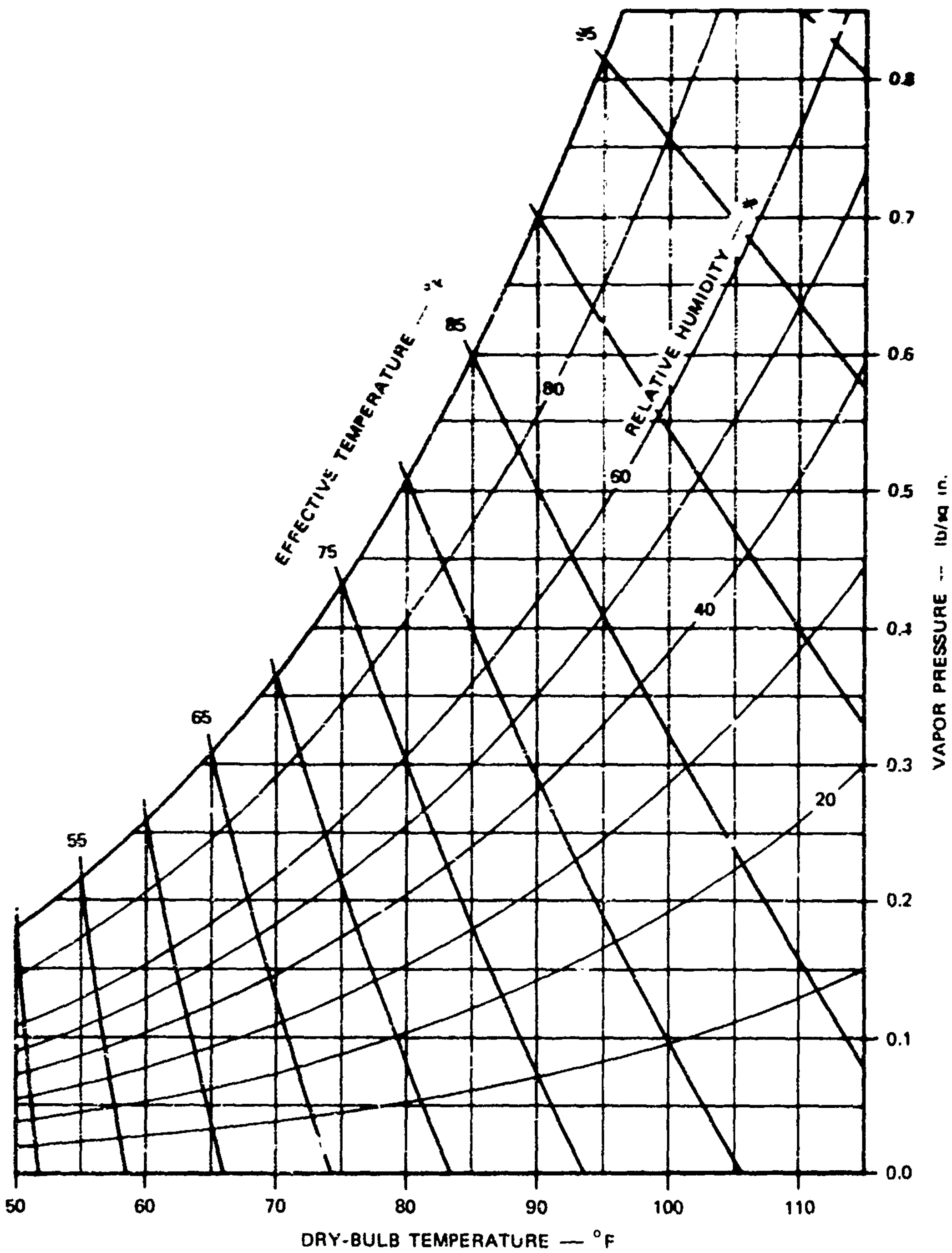

FIGURE 9 RELATIONSHIP OF EFFECTIVE TEMPERATURE TO DRY-BULB TEMPERATURE AND PARTIAL PRESSURE OF WATER VAPOR

S = slope coefficient for ET line (lb/sq in)/(°F).

The slope coefficient, S, is different for each discrete value of effective temperature, as shown graphically in Figure 10. Values for p_{wse} and t_e can be computed with Equation 65 or 66, or can be obtained from a table of pressures of saturated water vapor.[12] Equation 85 determines the end point of a ventilation process line for any value of effective temperature selected as a criterion for the shelter environment.

For use in psychrometric analyses, in conjunction with Equation 62, the equation for the effective temperature line can be expressed in terms of humidity ratio, W, rather than vapor pressure in moist air. Equation 85 then becomes

$$W = \frac{p_{wse} - S(t - t_e)}{p - p_{wse} + S(t - t_e)} \left(\frac{M_w}{M_a}\right) \tag{86}$$

in which p is the barometric pressure (lb/sq in).

Effects of Air Velocity

The foregoing discussion of effective temperature is based on assumptions that air velocities in shelter spaces will be low and that the effective temperature index associated with an air velocity of 20 feet per minute will be most applicable. Such a low velocity is comparable with local convection currents created by warm bodies in a space. Equation 85, Equation 86, Figure 9, Figure 10, and the effective temperature lines on Figure 8 all refer to this "still air" effective temperature at an air velocity of 20 feet per minute. Since drafts are usually considered objectionable, the "still air" effective temperature is the basic index used in air-conditioning practice. It has also been used extensively for evaluation of shelter environments.

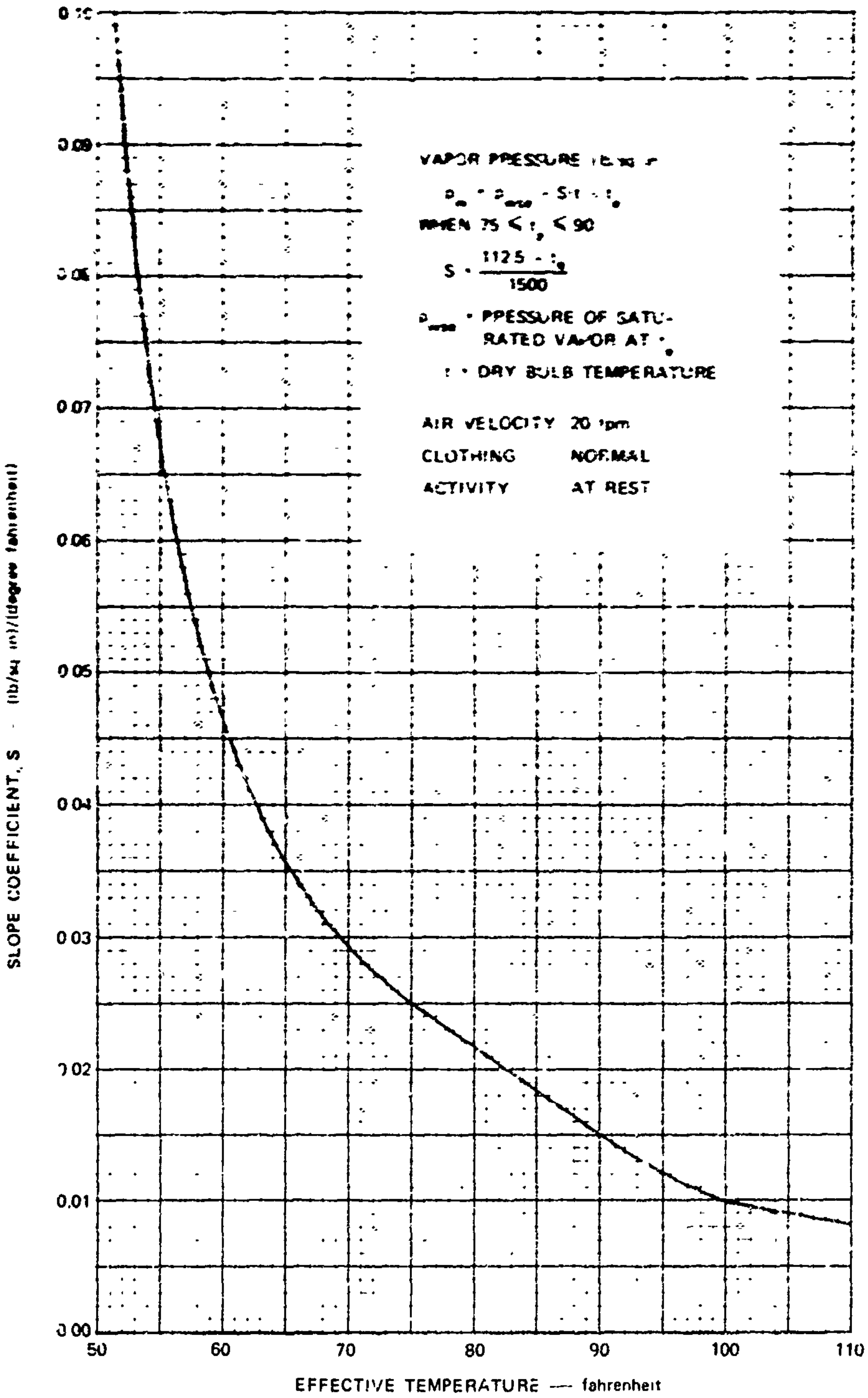

FIGURE 10 SLOPE OF EFFECTIVE TEMPERATURE LINES AS FUNCTIONS OF TEMPERATURE AND VAPOR PRESSURE

However, air velocities in ventilated shelters may be significantly greater than 20 feet per minute. The per capita ventilating rate needed to maintain a diurnal-average effective temperature of 82 FET or less during 90 percent of an average year is determined by probable climatic conditions at the shelter site. The required rate may range from 5 to 50 cfm per person.[30] For a large shelter, correspondingly large quantities may be needed, and air velocities may be sufficiently high to affect the criterion and consequently the requirement. For example, the Type AB composite shelter in Figure 1 and Table 1 has a capacity of 944 persons, if double deck bunks are used throughout. If the ventilating rate is 15 cfm per person and the pipe galleries are 7.5 feet in diameter, average air velocities based on gross cross-sectional areas would be about 43, 45 and 53 feet per minute, respectively, in the assembly room, dormitory room and dormitory galleries. In a more densely populated shelter or with a higher unit rate of ventilation, air velocities might be on the order of 100 feet per minute.

For a given combination of dry-bulb and wet-bulb temperatures, an increase in air velocity from 20 to 100 feet per minute appreciably lowers effective temperature. This is advantageous in a warm environment, but objectionable in a cool environment.

Alternative Indexes

An environmental index should identify equivalent psychrometric states and should take account of differences among individuals only to the extent that they affect heat transfer rates or heat transfer parameters. Differences in clothing, posture, and level of activity are significant in this respect, and, to accommodate such differences, a series of indexes may be better than one comprehensive index. An environmental index serves as the basis for rational environmental criteria. Duration of exposure, acclimatization, dehydration effects, and individual

differences such as age, sex, physique, susceptibility to thermal stress, state of health, metabolic disorders, and sweat sensitivity are all important considerations in establishing rational criteria. Thus, the environmental index is an intermediary between metabolic heat transfer and the environmental criterion.

Experiments on which the effective temperature index is based were necessarily of short duration, because the skin response rapidly becomes less sensitive to low rates of change in temperature or heat transfer. Deviations can be attributed to latent effects of moisture absorbed by clothing and, in hot environments, to the fact that the temperature of wet skin changes relatively little as rates of evaporation increase.[29]

The relative strain index is based on an analysis of heat transfer capabilities correlated with empirical data from many sources.[25] By definition, relative strain is the ratio of the evaporative cooling effect required for achieving thermal equilibrium to the maximum rate of evaporative cooling that can be maintained in a given environment. The original formulation is based on normally clothed subjects, a relative air velocity of 210 feet per minute and a metabolic rate of 100 kgCal/sq m (36.86 Btu/sq ft). Subsequent modifications have been made in part for application to nude persons, lower air velocities, and lower metabolic rates.[3] The slopes of relative strain lines can be compared with slopes of effective temperature lines on a psychrometric chart.[31] Although the two schemes are not based on the same conditions, the relative strain index tends to be more realistic in zones where the ET index is known to be in error and therefore represents a fundamental improvement. For application to shelters in which air velocities are low and sedentary occupants are wearing "optimum" clothing, a further modification of the scheme is needed.

Other recent work leading to development of a "comfort equation" is also based on the correlation of heat transfer capabilities with empirical

data.[27] The resultant equation is derived from a comprehensive analysis of the various heat transfer processes and has been correlated with experimental data to verify initial estimates of heat transfer coefficients, physiological parameters, and geometrical effects. This work represents a substantial advance toward derivation of a more consistent or definitive comfort index and could be extended to development of a generalized environmental index.

Metabolic Parameters

When comfort is the objective in design of an environmental control system, consideration is given to narrow ranges of temperature and humidity; ranges within which satisfactory comfort criteria and metabolic parameters have been established by experience as well as experiment. If the system is representative of good engineering practice, there will be sufficient inherent flexibility to accommodate probable deviations from design assumptions. Specific combinations of temperature and humidity are selected as criteria for both inside and outside conditions, and sensible and latent heat loads are calculated. These loads include sensible and latent heat gains caused by occupancy, and these are usually a relatively small part of the total load at design conditions. A uniform environmental state is obtained by distributing air or other heat transfer media in accordance with distribution of the loads. Air velocities in the space are low because air movement should not be perceptible.

When survival rather than comfort is the objective, and the system, by reason of dominant cost constraints, must be quite rudimentary, there is less guidance provided by experience, and the evaluation of parameters becomes more critical. For environmental analyses, the state of leaving air and slopes of the process line for air passing through a shelter are largely determined by these metabolic parameters.

For many years before 1963, data on metabolic parameters associated with a wide range of environmental states and levels of physical activity were published in graphical form for use by engineers.[32] With reference to normally clothed persons in an environment having low air velocities and equal dry-bulb and mean radiant temperatures, these data evaluated sensible heat losses as functions of dry-bulb temperature and total and latent heat losses as functions of effective temperature. The data did not differentiate between losses from skin surfaces and the respiratory tract, and sensible heat losses were not further partitioned into radiative and convective contributions. These graphical data on metabolic parameters have been deleted from later editions, but more complete information can be obtained from the original source.[33]

The original metabolic data relating to sedentary persons were derived from 267 tests in the same environmental chambers used for determining lines of equal comfort.[29] Environmental conditions included relative humidities of 20, 45, 70, and 96 percent; effective temperatures ranging from 44° to 100°F; and air velocities of "still air", 235 feet and 385 feet per minute. The test duration of four hours was preceded by a half-hour, preliminary adjustment period. The subjects were normally clothed, healthy college students of average physique from 19 to 24 years of age. Metabolic rates were determined by analysis of respiratory samples, latent heat losses were determined by weight losses on the assumption that all sweat was evaporated, heat storage was determined by changes in body temperature, and sensible heat losses were determined by difference. The environmental chambers were maintained at selected, psychrometric states, but did not serve as calorimeters. Even for similar test conditions, plotted data points are quite divergent.[26]

Metabolic parameters used in studies of shelter environments have been directly or indirectly derived from this source. For analytical purposes, the data have usually been approximated by linear or quadratic equations that are applicable within limited ranges. These data and data

from more recent investigations could well be evaluated, consolidated, adjusted in accordance with principles of heat-mass transfer, and adapted to critical conditions that would probably develop in shelters.

A shelter that is ventilated with unconditioned air from the ambient atmosphere would obviously tend to be warm and humid in summer and cool in winter. Since leaving air will have a higher temperature and humidity than entering air, there will be spatial as well as seasonal differences, and these may be large differences if the air is not well distributed. If there is no recirculation or air heating, all air entering the shelter will be virtually at outside conditions, which may be very cold in winter. It is reasonable to assume that shelter occupants would wear minimal clothing in a hot environment and heavy clothing in a cold environment and that sensations of discomfort would indicate appropriate clothing under any prevailing conditions. To accommodate the range of probable situations, a rationale is needed for evaluating metabolic parameters.

When air temperatures are more than about 96°F and little or no clothing is worn, evaporative cooling effects tend to hold mean skin temperatures at or near 95°F, unless ambient vapor pressures are excessive.[25] As air temperature decreases below 95°F, skin temperature also decreases, and discomfort is alleviated. After skin temperature decreases to about 92°F or the skin is no longer wet, a layer of light clothing would be added to prevent a further drop in skin temperature. Then, as air temperature continues to fall, additional layers of clothing would be added as necessary to maintain skin temperatures above the shivering threshold, which is 87 to 90°F for sedentary persons.[34] At any temperature, the amount of clothing should be optimum. This incremental procedure could apparently be continued to a low environmental temperature without an appreciable increase in metabolic rate, but health or other problems might be aggravated by prolonged environmental temperatures below 50°F.

Semi-empirical metabolic parameters have been evaluated for both nude and clothed persons in thermal equilibrium for environments having dry-bulb temperatures from 50° to 120°F and air velocities of either 30 or 100 feet per minute.[3] The assumed insulating value of clothing is 1.0 clo unit, which is the resistance associated with a surface-to-surface temperature difference of 0.88°F when heat flux is 1.0 (Btu/(hr)(sq ft). Figure 11 shows two exponential curves derived from these data for a sedentary subject with a constant metabolic rate of 400 Btu per hour. Equations for sensible and latent heat losses are also shown in Figure 11. These equations are of the form

$$q_S = a - be^{ct} \tag{87}$$

for sensible heat losses, and

$$q_L = 400 - q_S = A + Be^{Ct} \tag{88}$$

for latent heat losses, where t is the dry-bulb temperature of air in degrees Fahrenheit. Coefficients were determined in accordance with the following arbitrary constraints:

1. when $t = 0°F$, $q_S = 340$ Btu/hr
2. when $t = 95°F$, $q_S = 0$
3. when $t = 95°F$, $\frac{dq_S}{dt} = -20$

The constraints at 95°F were selected to approximate values for nude persons in the neighborhood of this temperature. The constraint at 0°F was selected to provide an assumed minimum value for latent heat,

$$q_{L(min)} = 400 - 340 = 60 \text{ Btu per hour.}$$

The curve for sensible heat intersects the reference data curve for clothed persons (1.0 clo) at about 69°F, which is a reasonable clothing requirement at this temperature. At lower temperatures, more clothing

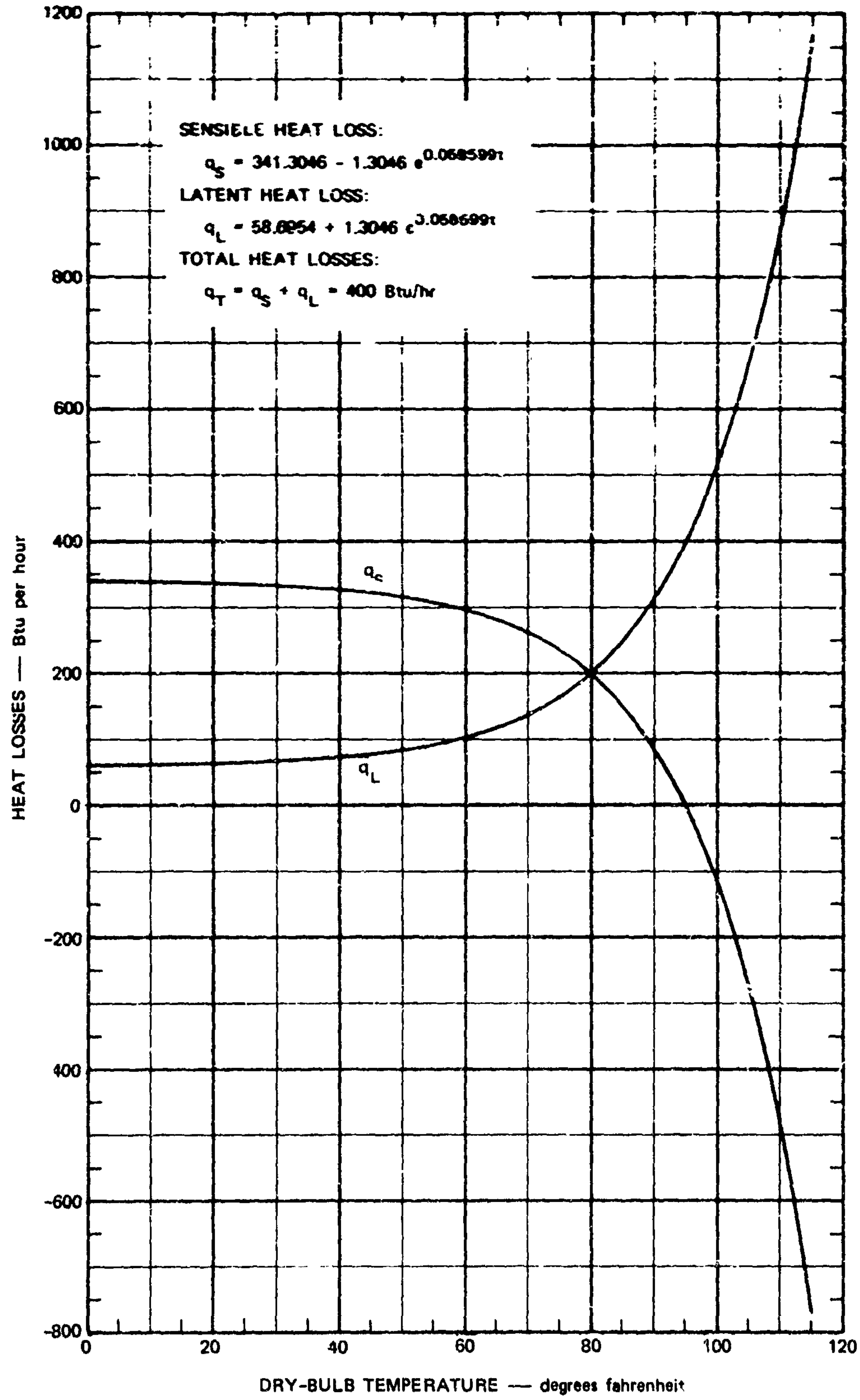

FIGURE 11 SENSIBLE AND LATENT HEAT LOSSES FROM A STANDARD PERSON IN THERMAL EQUILIBRIUM WHEN SEATED AT REST IN OPTIMUM CLOTHING

should be added incrementally to maintain a nearly constant skin temperature and metabolic rate. Values of latent heat losses at temperatures above 90°F may not be attainable if vapor pressure or relative humidity of the air is high. Figure 11 is presented not as parametric data but as illustrative of the kind of metabolic data that would be useful for generalized analysis of shelter environments. In Equations 17-19 and 32-35, $q_L = F(t)$. Further study and validation is needed to obtain a metabolic equation that can be used with confidence.

Metabolic Losses Due to Radiation

When total metabolic heat losses are partitioned, sensible and latent fractions are usually evaluated, but a further breakdown is rarely made because skin or clothing surface temperatures are not known. However, if principles of heat-mass transfer are used to determine metabolic parameters or to correlate empirical data, radiation effects must be considered. The sensible heat fraction includes both radiation and convection losses; at low air velocities, radiation effects tend to be the larger. Comparison of these two effects is facilitated by use of an equivalent radiation coefficient, h_r. If the Stefan-Boltzmann and Newton laws of thermal radiation are equated, for a body or object in an enclosure[25]

$$q_r = \sigma E A_0 (T_2^4 - T_1^4) = h_r E A_0 (T_2 - T_1) \quad (89)$$

where

q_r = net rate of radiant energy exchange,

T_2 = absolute temperature of one surface,

T_1 = absolute temperature of other surface,

h_r = equivalent radiation coefficient for black surface,

A_0 = effective surface area of body or object,

A_e = surface area of enclosure,

σ = Stefan-Boltzmann constant,

= 0.1713×10^{-8} (Btu)/(hr)(sq ft)(Deg R)^4

= 4.8776×10^{-8} (kgCal)/(hr)(sq m)(Deg K)^4

ϵ_0 = emissivity of object surface relative to black body,

ϵ_e = emissivity of enclosure surface relative to black body,

E = emissivity factor,

E = ϵ_0, if A_0 is small relative to A_e, and

$$E = \frac{\epsilon_0 \ \epsilon_e}{\epsilon_0 + \epsilon_e - \epsilon_0 \ \epsilon_e}, \text{ if } A_0 \text{ is almost as large as } A_e.$$

By solving Equation 89 for h_r and reducing, the equivalent radiation coefficient is,

$$h_r = \sigma T_1^{\,3}(R^3 + R^2 + R + 1) \qquad (90)$$

where $R = T_2/T_1$.

This is a cubic equation in R, the ratio of absolute surface temperatures. By repeatedly solving this equation for R with many values of h_r and T_1, data were obtained for Figure 12, which shows families of curves for the equivalent radiation coefficient, h_r, in terms of the two surface temperatures. The combined coefficient for heat transfer by radiation and convection is $h_{rc} = h_c + Eh_r$. More definitive information is also needed for evaluating the convection coefficient h_c. For free convection, h_c is dependent on both temperature and vapor pressure gradients in the neighborhood of skin surfaces. For forced convection, h_c is largely dependent on air velocity.[36]

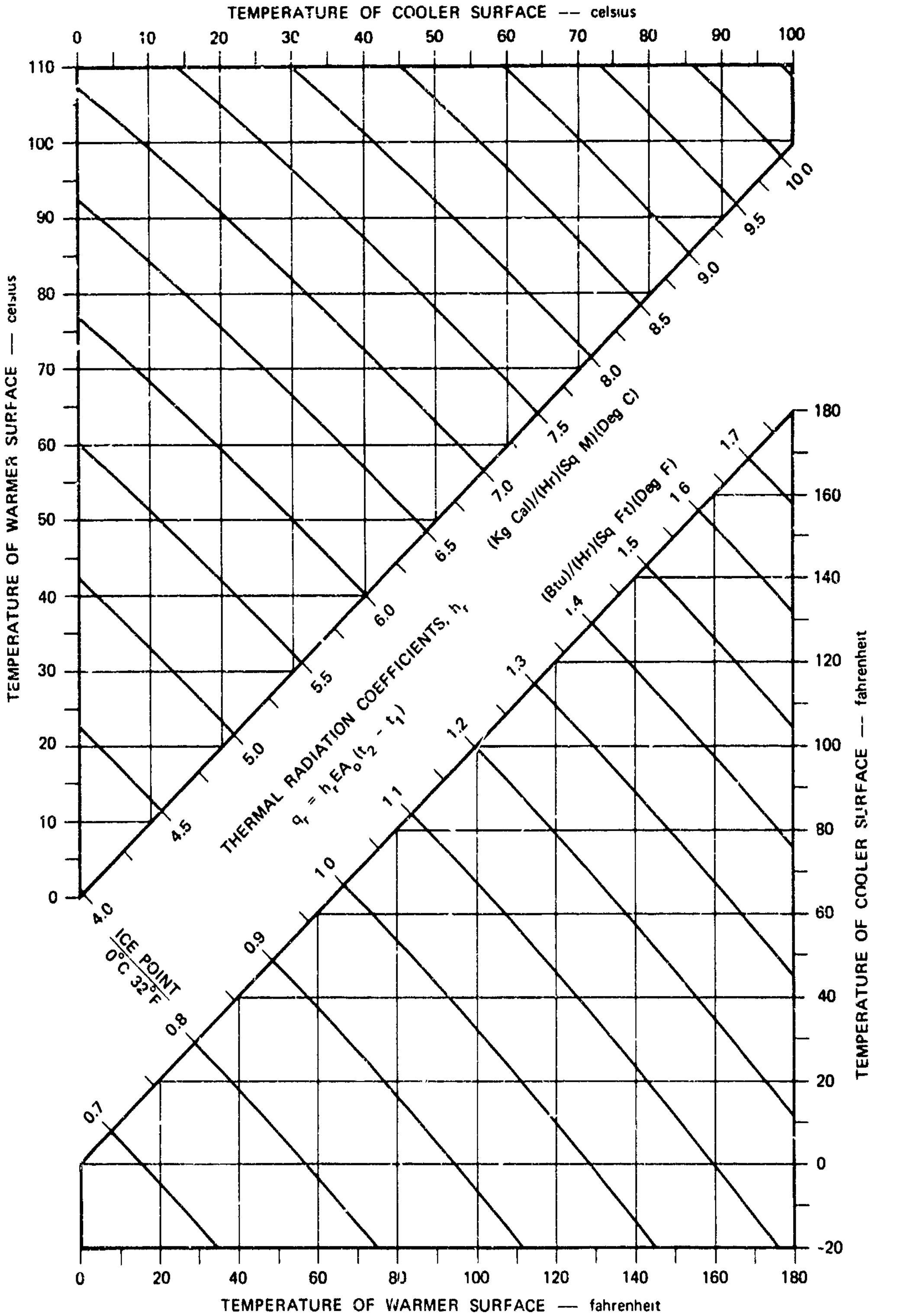

FIGURE 12 COEFFICIENTS FOR HEAT TRANSFER BY RADIATION BETWEEN AN OBJECT AND SURFACES OF ENCLOSURE

VI CONCLUSIONS AND RECOMMENDATIONS

Conclusions

The effective temperature index applies specifically to normally clothed sedentary individuals. Its applicability to atypical situations is questionable and subject to interpretation. The "still air" version of this index probably provides an adequate basis for shelter criteria in the range of 80 to 85 FET, when relative humidities are high. Criteria established on this basis would generally be conservative, if people wear minimal clothing or air velocities are relatively high, and would be severe for persons who are physically active. The index is not uniformly reliable in defining equivalent psychrometric states. It is not applicable to a cool or cold environment in which people would wear heavier clothing. If means are not provided to prevent cool or cold conditions, sufficient clothing is needed to maintain near-normal skin temperatures, to avoid cold stress, and to reduce the probability of widespread sickness.

More definitive data are needed with regard to metabolic parameters that cover the wide range of probable shelter environments, levels of activity, and associated clothing requirements. Available information can be consolidated and adjusted in accordance with principles of heat-mass transfer to improve reliability and to extend the range of metabolic parameters needed for analysis of shelter environments. The data should relate to conditions for thermal equilibrium with constant metabolic rates and should be normalized to represent a "standard" man with a skin surface area of about 20 square feet and dressed in clothing that is appropriate in the immediate environment. A linear correction may be adequate for nonstandard persons, but this should be verified. Metabolic equations for

latent heat losses are needed for sedentary and sleeping persons, when air velocities are about 20 and 100 feet per minute. In conjunction with these data or equations, limits imposed by high vapor pressures must be established, and the effects of changes in mean radiant temperature should be considered.

The equation of state for ideal gases constitutes an adequate and convenient basis for psychrometric analyses under any probable terrestrial conditions. Errors associated with real gas deviations and mixing interactions are small. Although a second degree approximation for enthalpy of saturated water vapor is more accurate, a first degree approximation is much more convenient and is considered adequate. Errors inherent in the analytical method tend to be obscured by uncertainties associated with metabolic parameters and environmental criteria. Molecular weights of dry air are increased appreciably by increases in carbon dioxide associated with low rates of ventilation. Wet-bulb temperature is not a convenient analytical parameter.

Adequate distribution of ventilating air can be obtained with minimal ductwork by using a shelter-system configuration in which favorable compartmentation causes air to flow uniformly through all spaces. Extremely cold areas can be largely avoided by recirculating a major proportion of relatively warm return air. These objectives could be attained in shelters derived from the composite plan shown in Figure 1. Comparative studies of seasonal and spatial variations in temperature and humidity, the performance of environmental control systems, the effects of heat conduction in contiguous masses of earth, and overall cost-effectiveness parameters can be based to advantage on shelter models with similar configurations.

The basic shelter system configuration includes an air intake facility in which there is a rock grille or packing that can be wet by water sprays,

as shown in Figure 2. This arrangement is potentially useful for attenuating weapons effects, for removing some air contaminants, and for air cooling with recirculated, well, or chilled water. A ventilation facility of this kind has a wide range of capabilities for environmental control and for future improvement. A similar facility for exhaust air could provide cooling water for an engine-generator or refrigerant condenser.

Recommendations

In general these studies are concerned with separate and combined effects of all factors that influence cost and performance of environmental control systems and with reciprocal effects of structure and system configuration on overall cost-effectiveness. These factors include spatial variations in the physical environment and performance of systems under winter as well as summer ambient conditions. For this purpose, reliable values of parameters must be used in the necessary analyses. This study emphasizes the need for improving and extending the range of metabolic parameters associated with optimum clothing and indicates the marginal applicability of the effective temperature index as a definition of equivalent psychrometric states. With regard to cool or cold environments, the immediate needs are for an index that relates clothing requirements to temperature and for associated metabolic data, particularly a consistent equation for latent heat losses. However, for application to warm conditions in shelters, the environmental index can well be modified in accordance with recent information. These interrelated problems should be studied further to obtain adequate metabolic data and improved bases for criteria. This information can then be used to prepare charts showing process lines for air traversing a shelter space and variations that would occur in the environment.

Plans of shelters should be prepared in more detail for use as models in studying the performance characteristics of environmental control systems and for evaluating overall cost-effectiveness factors. The configurations should be in substantial accordance with the composite plan outlined in this study. To determine the effects of various parameters, each shelter type could be represented by three shelter capacities, five variations of the basic system for environmental control, and three climatic locations.

An air intake facility that includes a wet rock grille, such as the concept outlined in this study, is potentially useful in shelters for attenuation of weapons effects and for cooling air with recirculated, well, or chilled water. A similar arrangement could also be installed in the stream of air leaving the shelter to provide cooling water for an engine generator or refrigerant condenser. Design studies should be made to evaluate these devices with regard to performance and compatibility.

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13. ABSTRACT

This investigation is preliminary to parametric studies concerned with (1) separate and combined effects of all factors that significantly affect costs and performance of environmental control systems for shelters and (2) reciprocal effects of structure and system configuration on overall cost-effectiveness. Representative configurations are outlined for shelters to be used as models in subsequent studies, which are oriented toward underground shelters that would provide protection from weapons effects in the 10-20 psi overpressure range. These configurations provide for recirculation and distribution of ventilating air with minimal ductwork. Alternative dormitory spaces of rectangular and gallery shapes are included; underground pipe galleries tend to enhance cooling effects due to heat conduction in contiguous earth. A concept is shown for a versatile air intake or exhaust facility incorporating a rock grille that could be wet with recirculated, well, or chilled water. A fundamental rationale is developed for analysis of nonuniform shelter environments, and deviations associated with use of ideal gas relationships, polynomial approximations, empirical equations, and variations in carbon dioxide concentration are evaluated. Derived values for equivalent thermal radiation coefficients are shown graphically. Environmental indexes, criteria, and metabolic parameters are examined with regard to adequacy and applicability to a concept for optimum clothing in shelters, and immediate needs for more definitive information are identified. (U)

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Wetted rock grilles						
Air washers						
Evaporative cooling						
Well water						
Chilled water						
Utility services						
Environmental analysis						
Psychrometric analysis						
Nonuniform environments						
Ideal gas laws						
Moist air properties						
Carbon dioxide						
Environmental indexes						
Environmental criteria						
Metabolic parameters						
Heat transfer						
Thermal radiation coefficients						